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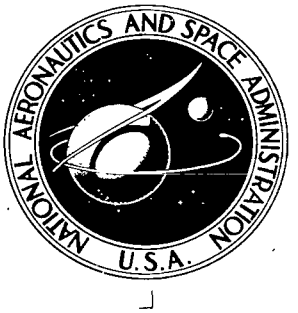

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AIRCRAFT CONTROL SYSTEM HYDRAULIC DRIVES

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V. G. Neyman

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16. Abstract Data are presented on the influence of aircraft and bench test hydraulic supply systems on the basic hydraulic drive characteristics. The characteristics of various test stand load simulation systems and their influence on hydraulic drive performance are discussed. Recommendations are given for selection of test stand hydraulic supply and load simulation systems which will provide the correct reproduction of the operating conditions. A technique is described for accelerated hydraulic drive service life testing, which is based on identification of weak elements and reproduction in the accelerated regime of the same degree of wear, aging, or fatigue damage accumulation of the weak elements as occurs under normal conditions.

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ANNOTATION

Questions of test bench and inspection testing of aircraft hydraulic drives are examined. Data are presented on the influence of aircraft and test bench hydraulic supply systems on the basic hydraulic drive characteristics. The characteristics of various test stand load simulation systems and their influence on hydraulic drive performance are discussed. Recommendations are given for selection of test stand hydraulic supply and load simulation systems which will provide the correct reproduction of the operating conditions. A technique is described for accelerated hydraulic drive service life testing, which is based on identification of weak elements and reproduction in the accelerated regime of the same degree of wear, aging, or fatigue damage accumulation of the weak elements as occurs under normal conditions.

The volume is intended for specialists involved in the development, testing, and operation of hydraulic drives. It may be useful also to university undergraduate and graduate students.

FOREWORD

The hydraulic drive has become the primary type of control system power drive in modern aviation. /3* While previously the hydraulic drive was used only in systems for manual or automatic control of aerodynamic surface position, on modern airplanes hydraulic drives are used in various electro-hydraulic and hydromechanical systems for controlling wing flaps, nosegear steering, air intake, wing sweep, and so on.

Along with broadening and increased complication of the hydraulic drive functions, there has been increasing complication of their operating conditions. Hydraulic drives often operate for long periods under conditions of high working fluid and surrounding medium temperatures — above 150° C. In many cases it is necessary to meet specified hydraulic drive characteristics at temperatures down to -20° C or lower, and also normal functioning and sealing at -60° C. The increase of flight speed and engine power has led to increase of the intensity of the mechanical effects on the hydraulic drives (vibration, shock, and linear accelerations).

Meeting the severe requirements on hydraulic drive reliability and service life under conditions of intense external mechanical and climatic effects is a serious problem.

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A large portion of the aircraft hydraulic drive operational failures are parametric failures, i.e., failures which show up in discrepancies between some output parameter and the specification value. The hydraulic drive output parameters defined by the engineering specifications are determined by both internal design-fabrication factors and external factors.

The design-fabrication factors include the geometric dimensions of the detail parts, clearances between adjoining detail parts, material properties, heat treatment, surface condition, characteristics of the mechanical, electromechanical, or hydraulic components which constitute the hydraulic drive, and so on. The design-fabrication factors and their tolerances are defined by the design documentation and the production process at the manufacturer's plant.

The hydraulic drive output parameters also depend on external factors, which for aircraft hydraulic drives may be divided into four groups:

1 — mechanical inputs; 2 — climatic inputs; 3 — output member loads; 4 — hydraulic supply system (hydraulic factors).

In order to build reliable hydraulic drives, we must know the external factors which arise during operation aboard the airplane, be able to evaluate their influence on the hydraulic drives, and take them into account during design.

In the book, we show that the hydraulic supply system may have considerable influence on the hydraulic drive characteristics. Undamped pressure oscillations which degrade hydraulic drive efficiency and reliability may arise in hydraulic supply systems with variable delivery pump. Recommendations are made to ensure stable operation of the hydraulic supply system and minimal influence of this system on the hydraulic drive characteristics.

Reliable hydraulic drive operation during service is usually verified during bench inspection tests, which are a very important stage in hydraulic drive development and production. In order to obtain an accurate evaluation during bench tests of whether the hydraulic drive meets the imposed requirements, it is necessary to simulate properly the actual external factors acting on the hydraulic drive in service. However, exact simulation of all the external factors during bench tests is impossible in practice. This is due to two basic reasons. The first is that the external factors and their combination are random quantities, which vary significantly both during each flight, and from flight to flight. We can pose the problem of simulating the external factors only during the time of some nominal typical flight. The second reason is the technical impossibility of building a test stand which permits simultaneous simulation of the control signals, load on the hydraulic drive output member, airplane hydraulic supply system, and also the entire combination of mechanical and climatic inputs. /5

This results in the necessity to simplify the bench tests, although it is still necessary to simulate as exactly as possible all the external factors which have significant influence on hydraulic drive technical characteristics and reliability.

In developing hydraulic drive test equipment and rational test methods, it is necessary to know the physics of the possible hydraulic drive failures due to the external factors. It is precisely this approach to the problem of justifying the rational scope of and technique for conducting bench tests that is taken in the present volume.

Although the hydraulic drive characteristics depend on the properties of the airplane hydraulic supply system, it is not possible in practice to reproduce the airplane hydraulic supply system completely in bench tests of the individual units. As a result of the analysis made herein of the influence of hydraulic supply system components (pump, regulating valves, hydraulic accumulators, working fluid properties, and so on), recommendations are given on how to

ensure, during bench tests, valid simulation of the airplane hydraulic supply system using a comparatively simple bench hydraulic system.

Under flight conditions, various loads act on the hydraulic drive output members: constant, position, position-dependent, inertial, and so on. Exact simulation of these loads is necessary during bench tests.

In many cases, the tests for resistance to heat and cold are the most complex and critical forms of hydraulic drive testing. Therefore, considerable attention is devoted in the book to analysis of the peculiarities of the techniques for these types of testing and the test equipment necessary to conduct such tests.

Hydraulic units of many different types are used in aircraft control systems. Hydromechanical servodrives (boosters) and electrohydraulic servocontrols with translational movement of the output member are most widely used. Therefore, in the book, our attention is directed primarily to analysis of hydraulic drives of these types. However, the problems examined also relate directly to the combined control units having manual and automatic electric control, and consisting of a combination of a booster and servocontrol, to electrohydraulic actuators, to hydraulic motor servodrives with rotational movement of the output member, and to other types of control system hydraulic units. /6

The author wishes to thank Yu. V. Tochilin and B. G. Turbin for their assistance in writing certain sections of the book, and the reviewer B. Ye. Berdichevskiy for his valuable suggestions made during review of the book.

The author will be glad to receive any comments and suggestions on the book, which should be addressed to Moscow B-78, 1st Basmannyy — per., House 3, "Mashinostroyeniye" Press.

CHAPTER I

BASIC CHARACTERISTICS OF AIRCRAFT CONTROL SYSTEM HYDRAULIC DRIVES

1.1. Hydromechanical Drives

The hydromechanical drives (boosters) are intended for aerodynamic /7
surface displacement, applied by the pilot through the control devices in the cockpit or by the airplane automatic control system. Boosters are also used in aerodynamic speed brakes, helicopter rotor pitch control systems, and in several other cases. Boosters connected in an irreversible scheme have found widest use in modern airplanes [14]. In these boosters, the load forces are not transmitted to the input member, and simulation of the load on the control organ from the action of the control surface hinge moments is accomplished with the aid of special springs or other more complicated mechanisms.

With regard to operating principle, the booster is a closed-loop servo system with position feedback, whose block diagram is shown in Figure 1.1. In this system, the slide valve displacement x and, consequently, the output member velocity are proportional to the difference between the input and output member positions.

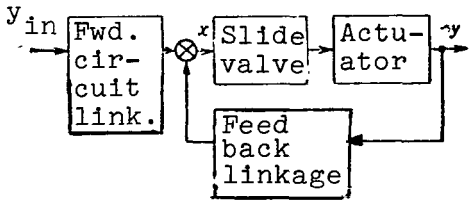

Figure 1.1. Block diagram of booster

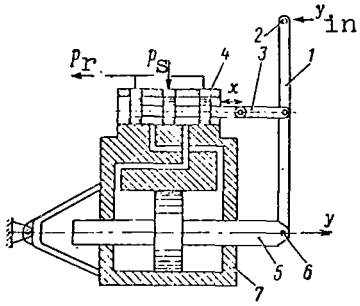

Figure 1.2. Booster schematic: /8
1 — lever; 2 — point of lever connection with control rod; 3 — link; 4 — distributor with cylindrical slide valve; 5 — power cylinder piston; 6 — point of piston rod connection with lever; 7 — power cylinder

The configuration of boosters may vary. Necessary elements of every booster are a slide valve type distributor and an actuator in the form of a power cylinder. Figure 1.2 shows a widely used booster configuration with cylindrical slide valve.

The input signal from the control stick and the feedback signal from the power cylinder piston rod are transmitted by the lever 1 to the slide valve. If the control rod displaces with the piston stationary, the lever rotates relative to the point of connection 6 and displaces the slide valve. As a result of slide valve displacement, fluid enters the corresponding cylinder chamber and displaces the piston until the slide valve is returned to the neutral position by the lever. The piston displacement is proportional to the rod deflection, and the coefficient of proportionality is determined by the dimensions of the lever arms between the points of hinged connection of the slide valve, control rod, and piston rod.

The distribution slide valve is the primary element of the /9
booster. Four-way slide valves of either cylindrical (Figure 1.2) or flat design are usually used in boosters. The constructional features of slide valves have been examined in detail in [3, 10, 11, 11, 28].

In one distribution device type with a plane slide valve (Figure 1.3), the annular slide valve of rectangular section displaces along an arc with rotation of the lever in the hole of which the slide valve is installed. The slide valve floats between a plate with input ports and a pad which is forced by a spring and input pressure against a support ring.

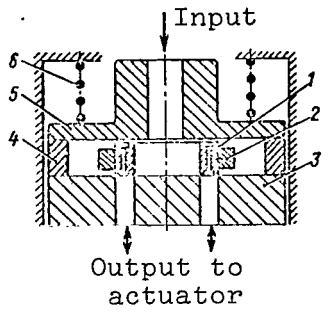

Figure 1.3. Schematic of distributor with plane slide valve:

1 — annular slide valve; 2 — rotating lever connected with control rotating shaft of distributor; 3 — plate; 4 — support ring; 5 — pad; 6 — spring

The clearance between the slide valve and the pad, determined by the relationship of the slide valve and ring heights, usually amounts to δ = 5 - 10 μ. The slide valve displacement force is reduced by introducing relieving ports in the slide valve. Plane slide valves are more reliable than cylindrical, since, if a mechanical particle gets into the working slot, the pad can lift and pass this particle, which makes it possible to avoid jamming of the slide valve. The plane slide valves make it easier to ensure the required working slot overlap, minimal clearance between the slide valve and the stationary elements (pad and plate) with which it is in contact, and minimal breakaway and displacement forces.

A power cylinder with symmetric piston (see Figure 1.2) is most often used as the booster actuator. In the case when the load is predominantly one-sided, power cylinders with differential piston are used, which permit reduction of the booster dimensions.

Booster construction and circuit differences are determined basically by the booster installation in the airplane and by the peculiarities of the control system scheme and the technique used to provide the required reliability. In order to improve control system reliability, multichannel (multichamber) boosters are used with two, three, and sometimes even four channels, with each channel supplied

from a separate airplane hydraulic system. The block diagram of a multichannel booster, which provides redundancy using the overpowering technique [4], is shown in Figure 1.4. The slide valves of all the channels are driven synchronously from a single control member, and the channel pistons are rigidly connected with one another. If one of the channels fails (for example, because of failure of one hydraulic system), the good channels provide the required output member displacement, overpowering the failed channel. Relief valves, which prevent pressure increase above the supply pressure, are installed in the power cylinders of each channel to prevent the occurrence of excessive pressures in the overpowering regime.

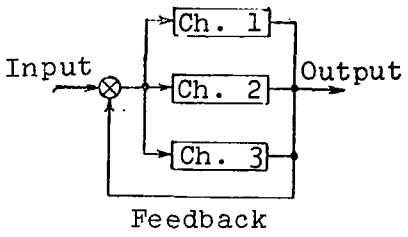

Figure 1.4. Block diagram of multichannel booster

/10

One of the basic booster characteristics is the static characteristic, i.e., the dependence of output member displacement on input member displacement. As a rule, this characteristic is nearly rectilinear; however, for boosters with complex forward loop and feedback kinematic linkage, the nonlinearity of the characteristic may be significant.

Usually, boosters have a limit on the output member travel. In this case, after the output member reaches the stop, the input member can displace by the magnitude of the error until reaching the distributor stop. The booster static characteristic is characterized by the transmission coefficient k_t, equal to the ratio of the output and input member displacements on the working segment of the characteristic. As a rule, the transmission coefficient k_t is close to one.

During booster bench tests, the following parameters associated with the static characteristic are checked:

— maximal input member travel ($2y_{in.max}$);

— maximal output member travel ($2y_{max}$);

— maximal error (y^*_{in}), i.e., the input member travel corresponding to total slide valve travel. Usually the maximal error is from 1 to 5% of the maximal input member travel.

These parameters are important for ensuring accuracy of booster kinematic linkage with the control organ in the cockpit and with the aerodynamic control surface.

A second important booster characteristic is the velocity characteristic, i.e., the dependence of the unloaded output member velocity v on error magnitude. The velocity characteristic is essentially the booster open-loop (without feedback) characteristic. This characteristic is shown in Figure 1.5. /11

The booster parameters associated with the velocity characteristic which are usually checked are:

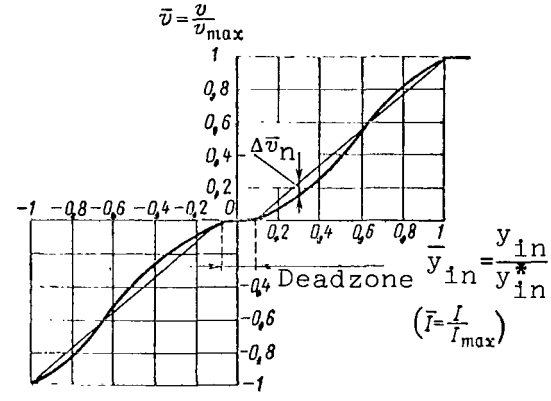

Figure 1.5. Booster velocity characteristic

— maximal output member velocity (sometimes the velocity of the output member with specified opposing load is also checked);

— difference of the maximal velocities, characterizing the asymmetry of the velocity characteristic;

— nonlinearity of the velocity characteristic;

— deadzone, i.e., the smallest input member displacement required to reverse the output member at the lowest possible velocity.

The first of these parameters defines the maximal airplane control surface deflection velocity, and the other parameters influence control system efficiency and stability.

Another important booster characteristic is the mechanical characteristic (Figure 1.6), which is the family of curves of output member velocity as a function of the load R for different constant errors. This characteristic is frequently described by a family of parabolas passing through the point of maximal developable force. However, the actual characteristics have somewhat less slope in comparison with the parabolas, because of the influence of passage resistance and leakage. /12

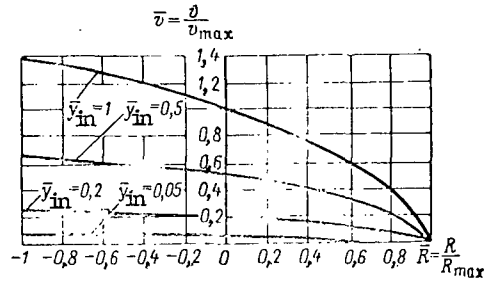

Figure 1.6. Booster mechanical characteristic

In the zone of small errors

($\bar{y}_{in} = \frac{y_{in}}{y^*_{in}} < 0.1$), the curves intersect the abscissa axis to the left of the point of maximal developable force ($\bar{R} = 1$), which is due to the influence of leakage in the slide valve pair.

Leakage has a marked influence on the flow and force static characteristics of the slide valve distributor, shown in Figure 1.7. These characteristics are curves of flow rate through the slide valve for zero load pressure and load pressure for zero flow rate versus slide valve displacement. The zone of leakage influence (linear zone of the force characteristic) is limited by slide valve displacement x_l, not exceeding 10% of the total travel. This zone is very important in analyzing the stability and static accuracy of hydraulic

drives. The slopes (k_{px} and k_{Qx}) of slide valve force and flow characteristics in this zone are usually found experimentally.

The maximal developable force, equal to the difference of the maximal hydraulic force on the piston and the friction force, determines the maximal hinge moment on the airplane surface which can be overcome. /13

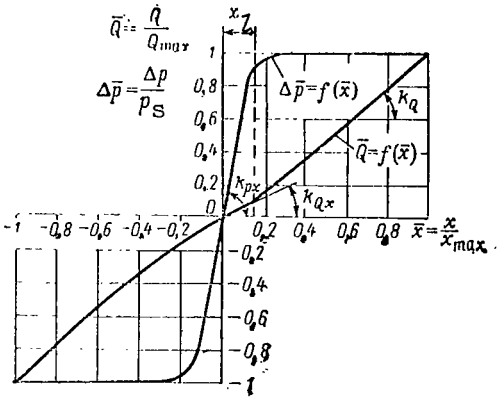

Figure 1.7. Slide valve static characteristics (flow and force)

The output member displacement resulting from applied load with stationary input member is very important for boosters (Figure 1.8). This characteristic defines booster static stiffness and, therefore, the surface control system stiffness. However, we note that the surface control system stiffness is also affected by the backlash and elasticity of the kinematic linkage from the booster output member to the surface and booster mounting stiffness. This characteristic has a hysteresis loop, due primarily to the influence of piston friction. The width of this loop is approximately equal to twice the friction force. The maximal force which can be held R_{he}, exceeding which leads to booster output member displacement to the stop, is equal to the sum of the hydraulic force on the piston and the friction force. Thus, the maximal force which can be held exceeds the maximal developable force by twice the piston friction force. The slope of the characteristic is determined primarily by the force characteristic of the booster distribution slide valve

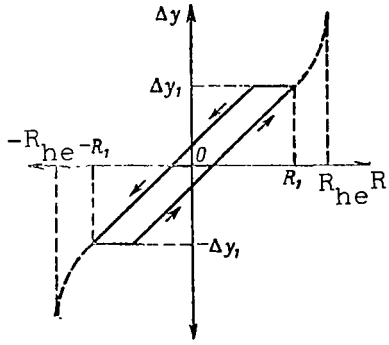

Figure 1.8. Booster output member displacement as function of load

(see Figure 1.7), since load on the output member causes, through the feedback, slide valve displacement such that the pressure differential across the booster piston equalizes this load. /14

The monitored booster parameters associated with the two characteristics indicated above are:

— maximal output member velocity for given opposing load;

— maximal developable force;

— output member displacement (Δy_1) for given load R_1 (see Figure 1.8).

The input member breakaway force is the monitored booster parameter which defines the required forces on the control organ in the cockpit or the force which can be developed by the servocontrol output element.

The internal leakage due to clearances in the slide valve pair has a significant influence on the average magnitude of the required flow rate, available useful flow rate, and working fluid temperature in the airplane hydraulic system.

An important booster characteristic is external sealing under all operating conditions. Usually, complete sealing is required for nonmoving joints, while for moving joints seepage of the working fluid without dripping is permitted. The severe requirements on hermeticity are due to the high danger of fire aboard an airplane, and also the rigid requirements of the airfield services with regard to cleanliness. However, sometimes a definite amount of leakage is permitted in the moving joints if organized leakage collection is provided on board the airplane.

Complete absence of working fluid leakage between the systems is required for the multichannel boosters in which each channel is supplied from an individual hydraulic system. /15

The requirements on booster and servocontrol dynamic characteristics are examined in Section 1.3.

1.2. Electrohydraulic Servocontrols

Electrohydraulic servocontrols (they are also sometimes termed servomechanisms) are used to control from the autopilot either the control surface position or the booster input member position. A schematic of one type of servocontrol is shown in Figure 1.9. The electrical control signal is applied to the windings of an electromechanical transformer, causing core displacement proportional to the control current. Recently, use has most frequently been made of the so-called "dry" transformers, which have improved reliability because of the absence of rubbing parts and operation of the windings in an air environment. In this type of transformer, the only part which extends into the working fluid is a flapper plate which is hermetically connected with a flexible tube and controls a hydraulic amplifier of the "nozzle-flapper and slide-valve" type. With the aid of orifices, the pressure differential at the ends of the cylindrical slide valve is altered as the flapper displaces and the slide valve displaces, until the hydraulic force is equalized by the force of synchronizing springs. Usually, four-way cylindrical slide valves of 5 - 8 mm diameter, having 0.2 - 0.5 mm travel, radial clearance δ = 3 - 6 μ, and overlap (1 - 3) δ, are used in servocontrols. The slide valve controls the working fluid flow rate through a power cylinder /16 with symmetric piston. One end of the piston rod is connected with the load, and the other is connected with a potentiometric or inductive feedback pickup. The electrical feedback signal, proportional

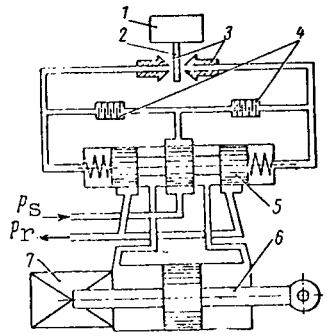

Figure 1.9. Servocontrol schematic:

1 — electromechanical transformer; 2 — flapper; 3 — nozzle; 4 — orifice; 5 — distribution slide-valve with synchronizing springs; 6 — power cylinder; 7 — feedback pickup

to piston displacement from the neutral position, is fed to the autopilot amplifier.

Multichannel servocontrols are used to increase control system reliability. Frequently, the multichannel units are constructed in the form of identical channels (modules) mounted side by side on a common bracket, and their piston rods are connected by hydraulic springs to a common output crossarm.

Redundancy and malfunction detection techniques and also block and functional diagrams of multichannel servocontrols are examined in [4].

The primary characteristics of servocontrols are the velocity and mechanical characteristics, analogous to the corresponding characteristics for boosters (see Figures 1.5 and 1.6). However, in contrast with boosters, the input signal for servocontrols is the control current I.

The primary monitored servocontrol parameters associated with these characteristics are:

— deadzone, i.e., the smallest control current change required for reversal of the unloaded output member of the lowest possible velocity;

— zero shift, i.e., the control current corresponding to the midpoint of the deadzone;

— magnitudes of the unloaded output member velocities for given control currents. Sometimes, only the velocity at the maximal control current is checked;

— asymmetry and nonlinearity of the velocity characteristic;

— output member velocity for given opposing load;

— force developed.

The influence of these parameters on flight vehicle performance is analogous to the influence of the corresponding booster characteristics examined previously.

The working fluid internal leakage, due both to leakage through the distribution slide valve and the hydraulic amplifier control flow, /17 is an important servocontrol parameter.

The alignment of the output member mechanical neutral with the feedback sensor electrical neutral is also checked for the servocontrols. This parameter (together with the zero shift) determines the accuracy of control surface setting in the neutral position.

The requirements on servocontrol sealing and leakage from one system into another (for multichannel units) are analogous to the requirements on booster sealing.

1.3. Hydraulic Drive Dynamic Characteristics

Strict requirements in regard to dynamic characteristics are usually imposed on aircraft hydraulic drives. Good dynamic characteristics are necessary for boosters and servocontrols in order to ensure control system stability and accuracy in all airplane flight regimes.

T. M. Bashti, N. S. Gaminina, V. A. Leshenko, V. A. Khokhlova, M. Giiona, and others,[2, 3, 10, 11, 13, 41] analyzed aircraft hydraulic drive dynamic characteristics. In order to evaluate the influence of external factors, we can describe the booster with inertial and position loads on the output member with adequate accuracy by the following system of differential equations:

$$\left.\begin{aligned} &k_3 x\sqrt{p_n - p_R \operatorname{sing} x} - k_{le} p_R = A_p \dot{y} + k_V \dot{p}_R, \\ &M\ddot{y} + f_y \dot{y} + C_p y = A_p p_R \\ &x = k_{in}\, y_{in} - k_{fb}\, y, \end{aligned}\right\} \qquad (1.1)$$

and the servocontrol with inertial load,

$$\left.\begin{aligned} &k_3 x\sqrt{p_n - p_R \operatorname{sing} x} - k_{le} p_R = A_p \dot{y} + k_V \dot{p}_R, \\ &M\ddot{y} + f_y \dot{y} = A_p p_R, \\ &k_{eha}\, I = T_{eha}\, \dot{x} + x, \end{aligned}\right\} \qquad (1.2)$$

where $k_3 = \sqrt{\frac{1}{\rho}}\mu b$; $k_V = \frac{V}{2E}$; M is the inertial load; f_y is the load damping coefficient; C_p is the slope of the position dependent load; p_s is the supply pressure; p_R is the load pressure; x is the slide valve displacement; μ is the slide valve slot flow coefficient; ρ is the working fluid density; k_{le} is the leakage coefficient; b is the total width of the rectangular slide valve ports; A_p is the active area of the power cylinder piston; V is the fluid volume in the power cylinder chamber; E is the elastic modulus of the working fluid; k_{in} is the transmission coefficient with respect to the input signal; k_{fb} is the feedback coefficient; k_{eha}, T_{eha} are the electrohydraulic amplifier gain and time constants. /18

In obtaining these equations, it was assumed that the return pressure is zero, there is no friction in the slide valve and piston, the slide valves have zero overlap, hydraulic drive mounting and its connection with the load are absolutely rigid, and the servocontrol electrohydraulic amplifier can be described by an aperiodic element [10].

The nonlinear expression for the flow rate Q through the slide valve slot in (1.1) and (1.2) can be linearized by expanding into a Taylor series in the vicinity of the steady-state motion regime (x_0,

Q_0, p_{R0}), and we obtain the following expression for the flow rate increment:

$$\Delta Q = k_Q \Delta x + k_p \Delta p_R, \tag{1.3}$$

where

$$k_Q = \left.\frac{\partial Q}{\partial x}\right|_{x_0,\, p_{R0}} = k_3 \sqrt{p_n - p_{R0}},$$

$$k_p = \left.\frac{\partial Q}{\partial p_R}\right|_{x_0,\, p_{R0}} = -\frac{k_3 x_0}{2\sqrt{p_n - p_{R0}}}.$$

In the zone of small slide valve displacements (for $|x_0| < x_l$) in (1.3), we can take, with account for leakage:

$$k_Q = k_{Qx} \quad \text{and} \quad k_p = \left.\frac{\partial Q / \partial x}{\partial p_R / \partial x}\right|_{\substack{x_v \to 0 \\ p_{R0} \to 0}} = \frac{K_{Qx}}{k_{px}}.$$

The linearized booster and servocontrol models (Figure 1.10), obtained with the aid of (1.1), (1.2), (1.3), make it possible to describe their dynamic properties approximately.

The booster and servocontrol transfer functions corresponding to these models can be obtained in the form /19

$$W_b(S) = \frac{y(S)}{y_{in}(S)} = \frac{k_{in}}{k_{fb}} \cdot \frac{1}{\dfrac{k_V M}{k_Q A_p k_{fb} + k_p C_p} S^3 + \dfrac{k_V f_y + k_p M}{k_Q A_p k_{fb} + k_p C_p} S^2 + \dfrac{k_V C_p + k_p f_y + A_p^2}{k_Q A_p k_{fb} + k_p C_p} s + 1}, \tag{1.4}$$

$$W_{sc}(S) = \frac{y(S)}{I(S)} = \frac{k_{eha} k_Q A_p}{A_p^2 + k_p f_y} \times$$

$$\times \frac{1}{S(T_{eha} S + 1)\left(\dfrac{k_V M}{A_p^2 + k_p f_y} s^2 + \dfrac{k_V f_y + k_p M}{A_p^2 + k_p f_y} S + 1\right)}. \tag{1.5}$$

Thus, the booster dynamic properties can be described approximately by a combination of amplifying, aperiodic, and oscillatory elements, where the latter two elements are obtained as a result of expanding the third-degree polynomial in the denominator of the transfer function. The servocontrol dynamic properties can be characterized by amplifying, integrating, aperiodic, and oscillatory elements.

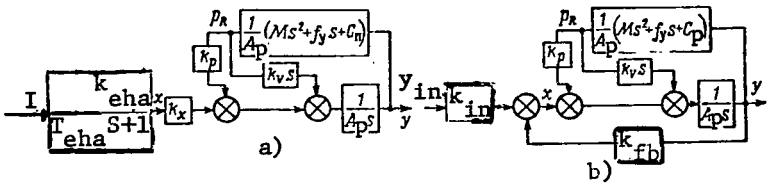

Figure 1.10. Block diagrams of linearized control servocontrol (a) and booster (b) models

Hydraulic drive dynamic properties are most often described by the frequency characteristics. Typical booster and servocontrol characteristics corresponding to the transfer functions (1.4) and (1.5) are shown in Figure 1.11. As is known, the amplitude-frequency and phase-frequency characteristics (AFC and PFC) show the ratio of the amplitudes and phase differences of the output and input signals as a function of constant-amplitude sinusoidal input signal frequency. Most often, the requirements on hydraulic drive dynamic characteristics are given in specifications in the form of the maximal permissible phase differences on the PFC and amplitude ratio range on the AFC for several input signal frequencies. /20

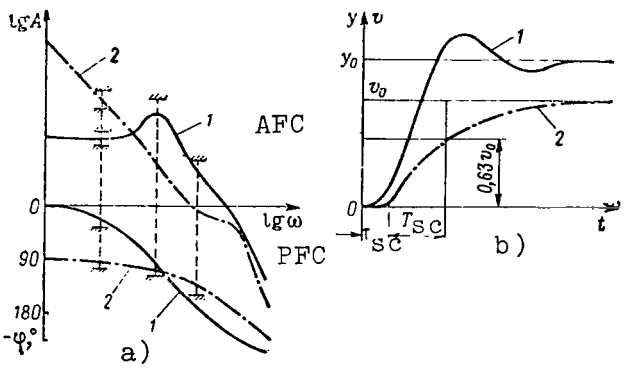

Figure 1.11. Typical dynamic characteristics of booster (1) and servocontrol (2):

a — frequency characteristics; b — transient processes

In many cases, it is necessary to account for hydraulic drive nonlinearities in the refined dynamic analysis. The basic hydraulic drive nonlinearities (other than the aforementioned relation for flow rate through the slide valve) are the limitation on flow rate and pressure in the supply line, friction in the slide valve, piston, and load, overlap in the slide valve pair, electromechanical transformer hysteresis (in the servocontrol), kinematic linkage backlash (in the booster), and so on.

The influence of electromechanical transformer hysteresis, slide valve pair overlap, friction, and backlash leads to the appearance of /21
a deadzone with respect to the input signal, which usually amounts to 3 - 5% of the maximal control current (or of the maximal error for the booster). The flow rate limitation causes saturation with respect to output member velocity. Therefore, in the case of control signals close to the deadzone or exceeding the signal which causes saturation with respect to output member velocity, the nonlinearities will distort markedly the frequency characteristics constructed on the basis of the linearized model. In order to account for the influence of the nonlinearities, the requirements on the frequency characteristics are sometimes specified for two or three input signal amplitudes, one of which corresponds to the linear hydraulic drive operation zone, the second corresponds to operation near the deadzone (usually around 0.2 I_{max}), the third corresponds to operation in the saturation zone.

The dynamic requirements are specified with a definite load on the output member (primarily the inertial load), in addition, the elasticity of the hydraulic drive mounting and the connecting linkage running from the hydraulic drive output member to the load are also sometimes specified.

For small inertial loads, when the time constant of the oscillatory element in the transfer function (1.5) is small, this oscillatory element, in accordance with the Kotel'nikov method, can be replaced by an equivalent lagging element. In this case, the

servocontrol transfer function can be written in the form:

$$W_{sc}(S) = \frac{k_{sc} e^{-\tau_{sc} S}}{S(1 + T_{sc} S)},$$

where k_{sc} is the gain, T_{sc} is the velocity rise time constant, τ_{sc} is the lag, characterizing the time from the instant of control current application until the instant of output member motion initiation (Figure 1.11b).

Accordingly, the dynamic requirements on servocontrols are sometimes specified in the form of the maximal permissible values of T_{sc} and τ_{sc}, with step-function control signal input.

Boosters are usually required to be dynamically stable, i.e., no auto-oscillations, vibrations, or output member travel irregularity with an inertial load under all operating conditions, with smooth movement throughout the entire velocity range and during sharp reversals. By auto-oscillations, we mean undamped oscillations of the output element with stationary input element; by vibrations, we mean slowly damped oscillations of the output member with considerable amplitude upon abrupt reversal of the input member; and by travel irregularity, we mean undamped oscillations of the output member in the case of input member motion with some constant velocity, or in some velocity range. Irregularity of travel is encountered most often in the "creeping" velocity range, where the influence of negative slope of the curve of friction force versus velocity [35] shows up to the greatest degree. /22

Of considerable importance for boosters are the dynamic characteristics with imposition of an output member load disturbance (with stationary input member). As is known, for a linear system the frequency characteristics and transient processes are analogous with disturbance application to the input and to the output load, and can be obtained one from the other by simple scaling. However,

because of the different influence of nonlinearities, the dynamic characteristics of real boosters may differ markedly for disturbance application to the input or to the output load.

Most often, the dynamic requirements with disturbance application to the booster output member are specified in the form of requirements on the dynamic stiffness, i.e., on the maximal ration of output element displacement amplitude to the amplitude of the sinusoidally varying load force at definite specified frequencies.

Thus, the requirements on hydraulic drive dynamic properties are specified in the following form:

— frequency characteristic (AFC and PFC) for given control signal amplitudes and frequencies;

— velocity rise time constant and lag (for the servocontrol);

— dynamic stiffness (for the booster);

— dynamic stability or absence of auto-oscillations, vibrations, and travel irregularity (for the booster).

1.4. Operational Requirements

The primary operational requirements on hydraulic drives are /23
reliability and service life. The basic concepts and terminology in the field of reliability are presented in GOST [Government Standard] 13377-67. The definition of reliability and service life is based on the concept of article operability. Operability is the state of an article in which it is capable of performing the specified functions with parameters established by the technical documentation requirements. The parameters characterizing function performance define the operational indices of the article. The technical documentations list the norms and permissible deviations for the established parameters, and also the external input level for

which the requirements, with respect to the parameters, must be satisfied.

An event amounting to disruption of article operability is considered to be a failure. The article state in which it does not meet one or more of the technical documentation requirements is termed a fault. We note that a significant portion of the faults do not lead to the appearance of failures.

The characteristics of article operability are service life (calendar duration of article operation prior to the limiting state specified in the technical documentation), operating time (duration of article operation in hours or other units), or life expectancy (operating time of the article until the limiting state specified in the technical documentation). The term life expectancy, expressed in hours of flight and ground operation, is most widely used for aircraft hydraulic drives.

According to the standard, reliability is the ability of the article to perform the specified functions while maintaining its operational characteristics in the specified limits for the required time interval or the required operating time. Reliability is a generalized characteristic, and includes freedom from failure, and longevity. By freedom from failure, we mean the ability of the article to remain operable in the course of some operating time without forced interruptions. The occurrence of failures during operation is not permitted for aircraft hydraulic drives, and, therefore, when evaluating freedom from failure, they may be considered nonrenewable /24 articles. If a fault develops, the hydraulic drive is not repaired, but rather replaced by another unit.

Reliable operation of a nonrenewable article can be characterized quantitatively by the following basic characteristics [41, 44]:

— probability P (t) of reliable operation in the limits of the specified time t = T (evaluates the probability that failure does not occur during the time period T under the specified operating conditions);

— failure probability $F(t) = 1 - P(t)$;

— failure probability density (distribution density) $f(t) = \frac{dF(t)}{dt} = -\frac{dP(t)}{dt}$;

— failure rate $\lambda(t)$ (probability of failure per unit time after a given instant of time);

— mean time of reliable operation T_0;

— gamma-percent life expectancy T_γ (time in the course of which the established failure-free operation probability $P(t) = \gamma\%$) is guaranteed.

The dependence of these reliability characteristics on time and their interconnection are defined by the failure distribution law. Exponential and normal distributions are most frequently used to evaluate the reliability of aircraft hydraulic drives.

The exponential distribution corresponds to the appearance of only random failures, the reason for which cannot be associated with the slow damage accumulation processes in components of the article. Such failures are the consequence of a combination of unfavorable factors: the presence of production defects, violation of operating rules, occurrence of off-design operating regimes or the worst extremal combination of several parameters which are not considered during design. Random failure occurrence is independent of article operating time. The exponential distribution can be characterized by the failure rate λ, which is constant for given operating conditions. The other reliability characteristics are easily expressed in terms of λ [44]. The dependences of the basic reliability characteristics on operating time for the exponential distribution are shown in Figure 1.12.

The normal distribution corresponds to the occurrence of progressive (wearout) failures, associated with slow damage accumulation /25

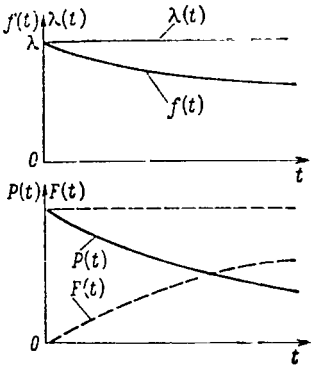

Figure 1.12. Graph of basic reliability indices versus operating time for exponential distribution

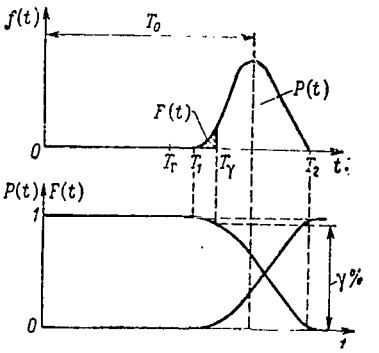

Figure 1.13. Graph of basic reliability indices versus operating time for normal distribution

(wear, aging, fatigue damage accumulation, and so on), and also parametric failures [41]. The normal distribution can be described by two parameters: the mean time T_0 of reliable operation and the dispersion D (or mean square deviation $\sigma = \sqrt{D}$). The quantity D characterizes the scatter of the operating-time-to-failure relative to the mean value T_0. The larger D, the larger the scatter of the operating-time-to-failure relative to T_0. All the other reliability characteristics can be expressed in terms of the parameters T_0 and D (or σ [44]. The dependences of the basic reliability characteristics on operating time for the normal distribution are shown in Figure 1.13. The scatter of the operating-time-to-failure is due to the different initial state of the components of the article, which depends on a random combination of design and fabrication errors and different article operating conditions.

If an article is subject to the combined action of random and progressive failures, the probability of its reliable operation is determined by multiplying the probabilities of reliable operation with random failures $P_1(t)$, which obey the exponential law, and with progressive failures $P_2(t)$, which obey the normal law /26

$$P_\Sigma(t) = P_1(t) P_2(t).$$

If the exponential and normal distribution parameters λ, T_0, and σ are known, we can easily determine the quantity $P_\Sigma(t)$. In the case of combined action of random and progressive failures, in the initial article operating period the random failures play the dominant role, and then with increase of the operating time the progressive failures have increasing influence.

The most widely used reliability characteristic for aircraft hydraulic drives is the probability of reliable operation during flight. An indirect reliability characteristic is the guaranteed operating time; this is the time during which the manufacturer guarantees performance by the article of all the established requirements if the user observes the operating rules. Complete reliability is usually required in the course of the guaranteed operating time, in this case random failures due to improper operation (improper connection of the hydraulic and electrical supply, occurrence of off-design operating regimes with respect to pressure, temperature, or load, excessive contamination of the working fluid, presence of moisture in the working fluid, and so on) are not considered the fault of the manufacturer, and are not considered in evaluating hydraulic drive reliability. During the guaranteed operating time, performance of the required inspection operations (periodic filter cleaning, bearing lubrication, and so on) is mandatory. Since it is economically advantageous to the manufacturer to increase the guaranteed operating time to the maximum degree possible, in the course of time a guaranteed operating time close to T_1 is established (Figure 1.13), which permits considering the guaranteed operating time an indirect hydraulic drive reliability characteristic.

In experimental production and the beginning of series production, random failures occur during article tests or operation as a result of design errors, improperly specified article operating conditions, improper article servicing because of inadequate preparation of the servicing personnel while learning the new equipment, /27
and also the appearance of latent production defects. In order to

minimize the probability of random failures, it is necessary to carry out extensive technical and organizational measures during design, production, and operation. During design, special attention should be paid to proper specification of the operating conditions, loads, and external inputs, and also to accounting for the broad range of characteristic reliability requirements [6]. In the production stage, it is necessary to perform careful inspection in all stages of the production cycle. In order to reduce the influence of random failures due to latent production defects, it is advisable to test run each unit at the manufacturer's plant. During operation, it is necessary to ensure strict observation of all the requirements of the operating instructions and other governing documents, and also to improve the qualification of the servicing personnel in every possible way. The occurrence of random failures in operation can be practically eliminated for aircraft hydraulic drives if all these measures are followed.

However, it should be noted that in the hydraulic drive design stage, it is well to make calculations of the probability of failure-free operation during flight on the basis of the exponential hydraulic drive component failure distribution law in order to obtain a comparative evaluation of the freedom from failure of several system variants.

The reliability of aircraft hydraulic drives is characterized not only by freedom from failure, but also by longevity, by which we mean the ability of the article to remain serviceable to the limiting state with the required interruptions in operation for technical servicing and overhaul. The longevity of aircraft hydraulic drives is characterized by the technical (specified) operating time, i.e., by the total article operating time, upon reaching which, operation must be terminated regardless of article condition. At the technical operating time, the hydraulic drive is overhauled with replacement of parts and components (for example, seals, feedback potentiometers, and so on) which have the shortest service life. The technical

service life is established with account for safety, economy, and obsolescence.

In many cases, the operational conditions for aircraft hydraulic drives are, on the average, less severe than the conditions specified in the technical documentation. This leads to a situation in which the hydraulic drives, after operating in service for the guaranteed operating time, can still remain serviceable for a considerable time. Therefore, the concept of operating time to first overhaul is introduced, which exceeds the guaranteed operating time and is established on the basis of joint decision of the manufacturer and user. /28

CHAPTER II

AIRCRAFT CONTROL SYSTEM HYDRAULIC DRIVE OPERATING AND TEST CONDITIONS

2.1. Classification of External Factors

The hydraulic drive output parameters are determined by the combination of internal design-technological factors (a_i), and the external factors (b_j). The internal factors determine the state of the system elements and components, and characterize their geometric dimensions, material properties, detail part surface finish, the mechanical, hydraulic, and electromechanical characteristics of the elements, and so on. It is convenient to divide the external factors for hydraulic drives into four basic groups: 1 — mechanical inputs (b_1); 2 — climatic inputs (b_2); 3 — output member load (b_3); 4 — hydraulic supply system (b_4 — hydraulic factors). The forms of external factors acting on aircraft hydraulic drives are shown in Figure 2.1. /29

The magnitude of the generalized hydraulic drive parameter w can be written in the form:

$$w=\varphi[a_i(t);\ b_{1j}(t);\ b_{2j}(t);\ b_{3j}(t);\ b_{4j}(t)]. \qquad (2.1)$$

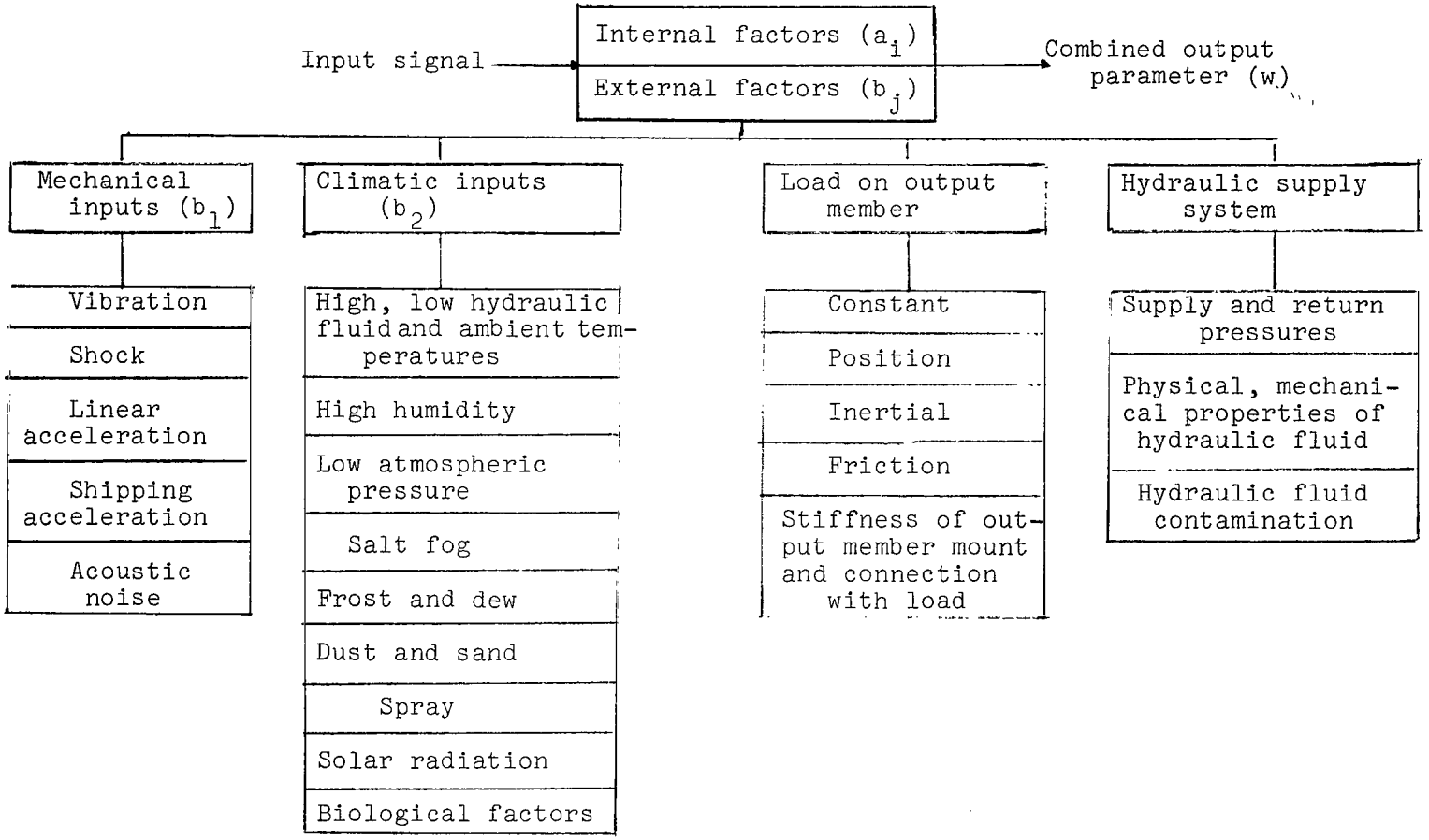

Figure 2.1 Diagram of factors acting on aircraft hydraulic drives

The hydraulic drive parameters in the initial state under normal conditions (i.e., under normal climatic conditions and absence of mechanical inputs) are defined by the expression:

$$w_0 = \varphi[a_{i0},\ b_{2j0},\ b_{3j0},\ b_{4j0}], \tag{2.2}$$

where b_{3j0} and b_{4j0} are the nominal vlaues specified in the technical documentation, characterizing the load on the hydraulic drive output member and the hydraulic supply system parameters.

Thus, the magnitude of the parameter w_0 is determined by a random combination of design and technological errors (Δa_{i0}), equal to the difference between the actual magnitudes of the design and technological parameters (a_{i0}) and their nominal values (a^*_{i0}). The accuracy with which b_{2j0}, b_{3j0}, b_{4j0} are maintained may also have some influence on w_0. /31

In the process of hydraulic drive operation, the quantity w_0 changes, and the changes Δw consist of the reversible changes $\Delta w'$ due to the action of the external factors and the irreversible changes $\Delta w''$ caused by the accumulation of permanent damage (wear, aging, and so on), and by the variation of the quantities a_i.

After linearizing (2.1), we obtain:

$$\Delta w = \Delta w' + \Delta w'',$$

where

$$\Delta w' = \sum_{i=1}^{n}\sum_{j=1}^{m}\frac{\partial \varphi}{\partial a_i}\frac{\partial a_i}{\partial b_j}\Delta b_j + \sum_{j=1}^{m}\frac{\partial \varphi}{\partial b_j}\Delta b_j; \tag{2.3}$$

$$\Delta w'' = \sum_{i=1}^{n}\int_0^{t_0}\frac{d\varphi}{da_i}\left[\theta_i(t) + \sum\frac{\partial \theta_i}{\partial b_j}\Delta b_j\right]dt. \tag{2.4}$$

Here, $\theta_i(t) = \frac{\partial a_i}{\partial t}$ is the rate of the irreversible changes of a_i under normal conditions, which is a continuous random quantity. Since the damage accumulation process is irreversible, θ (t) > 0.

We see from (2.3) that the external factors can influence the reversible parameter changes either directly (for example, influence of temperature-dependent viscosity changes on leakage flow rate) or through changes of the design and technological factors a_i (for example, influence of temperature-dependent change of structural element dimensions on output member velocity).

The degree of influence of the external factors on the design and technological factors and on the output parameters, and also the degree of influence of change of the design and technological factors on the output parameters is characterized by the weighting coefficients $\frac{\partial a_i}{\partial b_j}$, $\frac{\partial \phi}{\partial b_j}$, $\frac{\partial \phi}{\partial a_i}$.

The magnitudes of the hydraulic drive parameter changes during operation depend on the random variation of the magnitudes of the external factors during flight Δb_j (t), and on their worst combination. Only certain of the design-technological and external factors affect each of the output parameters. /32

The influence of the design and technological factors and their errors on hydraulic drive parameters has been analyzed quite thoroughly in several studies [3, 11, 13, 28, 42, and others]. Therefore, it is advisable to examine in more detail the influence of the external factors on hydraulic drive parameters.

The action of the various external factors may either lead only to reversible changes of certain hydraulic drive output parameters (b_j'), or may cause accumulation of irreversible changes (b_j'').

It is also advisable to associate with the factors b_j'', those external factors which influence the accumulation of fatigue damage leading to hydraulic drive structural element fatigue failure. The factors b_j' can be considered to be those which determine hydraulic drive operating stability (i.e., retention of the drive parameters

within the required limits during action of the external factors), while the drive factors b_j'' are those which determine hydraulic drive durability (or strength), i.e., retention of parameters (in the required limits) after long-term action of the external factors.

The external factors affect only the stability, and have no influence on the durability ($\Delta w'' \approx 0$), if the corresponding weighting coefficients $\frac{\partial \phi}{\partial a_i}$ and $\frac{\partial \theta_i}{\partial b_j}$ are very small or the external factors act for a short time. For those external factors which influence both hydraulic drive stability and durability, the influence on stability should be evaluated on the basis of the maximal magnitudes of the factors during operation, and the influence on durability is evaluated on the basis of their mean values.

It should be pointed out that the inputs which take place not only during hydraulic drive operation on the ground and in flight, but also during their storage and transportation, are included among the external mechanical and climatic inputs.

2.2. Mechanical Inputs

The primary form of mechanical input to aircraft hydraulic drives are intense vibrations, which depend on the airplane type, its flight conditions, and the location of the hydraulic unit on the airplane. The primary source of vibrations aboard the airplane is the engine, which creates a broad vibration spectrum as a result of instability of the combustion process in the combustion chamber, and also static or dynamic unbalance of the engine rotor. The most intense vibrations arise at the frequency corresponding to the engine rotor rotational speed. Another vibration source is the action of the variable aerodynamic forces which arise as the air flows around the airplane. The vibrations due to the action of these aerodynamic forces on the fuselage, wings, and other airplane components are random. Vibrations also arise under the influence of ground surface

/32

reaction forces, which cause elastic vibrations of the structure during taxiing, takeoff, and landing. The last vibration source is acoustic noise, created primarily by the engine [14].

The vibrations acting on aircraft equipment have a three-dimensional nature and constitute a random-periodic process, with the power of the periodic components of the vibration being comparatively small.

The experimental determination of data on vibrational inputs, and the technique for establishing the requirements in regard to vibrational inputs for the onboard equipment are very complex problems. Usually, all aircraft are broken down into several groups based on mission and powerplant type in order to obtain these requirements. In each group, the airplane is divided into several basic zones [14]. On the basis of correlation of the vibration measurement results during flight tests of airplanes of the corresponding groups, we determine for each zone the requirements with respect to frequency range, and also the vibration spectral density, amplitudes, or accelerations.

For a typical case, the aircraft instrument vibration frequency range varies from 10 to 200 Hz, the vibration amplitude magnitude does not exceed 1 mm, and the vibrational acceleration does not exceed 5 g [33]. In certain cases, for example, in the U.S.A. standard /34
MIL-STD-810B for onboard equipment, broadening of the frequency range to 500 or 2000 Hz and increase of the vibrational acceleration to 10 g are provided for.

We should point out that the questions of vibrational input standardization are very complex, and at the present time there is no unified technically rational approach to the solution of these questions. The technique most widely used is simulation of the actual vibrations with the aid of harmonic vibration; however, the equipment operating regime in this case is usually considerably more severe than the operating conditions. The harmonic vibrations are

represented either in the form of a combination of several regimes with specified fixed or resonant frequencies, or in the form of several frequency ranges in which the vibrational frequency is varied smoothly at a given rate. The specific norms for vibrational inputs are intimately related with the vibration test methods, and will be examined in more detail in Chapter VI.

The second form of mechanical input are shocks, which may arise during airplane takeoff and landing, when firing from onboard weapons, nearby explosion of enemy shells, and so on. An example of the requirements with respect to shock inputs would be exposure to a total of 10,000 shocks with maximal acceleration 5 g, with shock pulse duration 20 - 50 ms [33].

During airplane maneuvering, acceleration, and deceleration, linear (centrifugal) accelerations act on the equipment. As an example of the norms on linear acceleration, we can cite the value 6 g for heavy airplanes, and 9 g for light airplanes [33]. In certain standards (for example, in MIL-STD-810B), the linear acceleration norms are specified differently, depending on the orientation of the unit and the zone in which the unit is located aboard the airplane.

Intense acoustic noise acts on airborne equipment; however, this noise has practically no effect on hydraulic drives.

The accelerations which arise during shipping have effects which are very hazardous for hydraulic units. In certain cases, these accelerations may exceed those which arise during operation. The effect of the accelerations which arise during shipping are often replaced by the equivalent vibrational or shock inputs. /35

Of the mechanical inputs noted above, the vibrations affect both hydraulic drive strength and stability, and shocks and accelerations which arise during shipping affect only hydraulic drive strength, and the linear accelerations affect only hydraulic drive stability.

2.3. Climatic Inputs

Changes of the ambient and hydraulic fluid temperatures have the largest influence on hydraulic drives. The minimal ambient and fluid temperatures, which can occur with the airplane parked in the polar regions, reach -60° C. During airplane flight at high altitudes, negative temperature of the hydraulic fluid (down to -20 - -40° C) may occur for long periods because of the low ambient air temperatures (Figure 2.2) . The maximal ambient and working fluid temperatures for hydraulic drives of the modern supersonic airplane may reach +150° C, and, in certain cases, even higher values. /36

Aircraft hydraulic drives operate under low atmospheric pressure conditions. The curve of atmospheric pressure versus altitude above sea level is shown in Figure 2.2. The standard magnitude of the reduced atmospheric pressure corresponds to the maximal airplane flight altitude. For example, for maximal altitude 26 km, the pressure is 0.2 N/cm^2 (15 mm Hg).

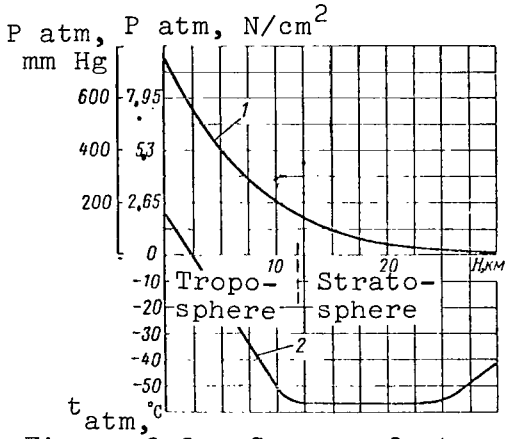

Figure 2.2. Curves of atmospheric temperature (2) and pressure (1) versus flight altitude

An important climatic factor is high ambient air humidity. The high humidity norms are equivalent to long-term action of moisture under operating conditions. These norms are insufficiently justified at the present time, and long-term experiments are required to establish more rational norms. As an example, we can cite the continuous action of moisture for 10 or 21 days with relative humidity 95% and temperature 40° C, or the cyclic action of moisture for 9 or 21 days (cycles) with maximal temperature 40° C and relative humidity 95% (GOST [Government Standard] 15151-69).

The frost and dew which form while parked or in flight act on the hydraulic drives. These conditions can occur during airplane startup after extensive exposure to negative temperatures, and also when an airplane, with temperature close to 0° C flies at high speeds in clouds, fog, or above a precipitation band.

Salt (ocean) fog may act on the hydraulic drives of airplanes based at coastal airfields. The action of ocean fog extends to altitudes of 500 meters and inland to a distance of 9 - 20 km [25].

The hydraulic units located in unprotected areas of the airplane (for example, on the landing gear strut) are subjected to the action of dust and sand, solar radiation, and water spray. The norms for these inputs are presented in GOST 15151-69. Biological media (mold fungus, microorganisms, insects, and rodents) also act on hydraulic drives. As a rule, the effects of these media on the hydraulic units are not standardized, and resistance of the system to their action is guaranteed by the components, materials, and coatings used (GOST 15963-70). /37

Of the considered climatic inputs, the reduced atmospheric pressure, and also the frost and dew, affect only hydraulic drive stability,(i.e., they do not lead to irreversible changes). The action of the remaining factors can influence hydraulic drive durability (i.e., it may lead to irreversible changes).

We should note that the working fluid temperature is a hydraulic factor, but from the methodological viewpoint it is more conveniently considered in the climatic input group since, together with the ambient air temperature (and to a considerably greater degree), it determines the hydraulic drive thermal operating regime.

2.4. Operating Regimes

Other conditions being the same, the hydraulic drive operating regime is characterized by the control signal and output member load,

which are determined by the hydraulic drive function aboard the airplane. The real control dignals are approximated quite well by two types of signals — sinusoidal and step function. A nearly sinusoidal control signal is characteristic for boosters and servocontrol operating in the airplane stabilization mode, while a step-function signal is characteristic for the short-term periodic-action boosters (for example, those which actuate speedbrakes). A step-function control signal may also act on the hydraulic drives during change of airplane flight regime. It should be noted that, in practice (particularly for boosters), the control signal is not an ideal step function; the signal rise time amounts to from a few thousandths of a second to several tenths of a second.

A combination of loads which differ in nature usually acts on the hydraulic drive output members. The primary load type for boosters is the hinge moment (positional or spring loading) which arises from the aerodynamic action of the air stream on the deflected control surface. This load can be considered to vary in direct proportion to the piston rod displacement relative to the neutral position. A second load type is the inertial load, which is due to the influence of the moment of inertia or mass of the control member being displaced (control surface, spoiler, air intake doors, and so on). This form of load is usually specified in the form of an inertial mass referred to the hydraulic drive piston rod. Along with the magnitude of the referred mass, in many cases the stiffness of the hydraulic drive mounting and the elasticity of its connection with the control member are specified, which are important for hydraulic drive dynamic stability. /38

A constant unidirectional load, arising from an unbalanced aerodynamic force or moment on the control system actuating member, may also act on the hydraulic drives. The last load type is the friction which arises in the actuating member and in the connection between the hydraulic drive output member and this member. In regard to its nature, the friction is usually nearly dry (or Coulomb).

This group of external factors may influence both hydraulic drive stability and strength.

2.5. Aircraft Hydraulic Supply Systems

In most cases, when analyzing hydraulic drives, the supply (delivery) and return pressures are considered constant, and the influence of changes of the physical and mechanical properties and state of the working fluid is not considered. Such a hydraulic supply system can be termed "ideal". In real hydraulic systems, the variability of the supply and return pressures, working fluid contamination, amount of air in the fluid and the change of its physical and mechanical properties may have significant influence on the hydraulic drives. The hydraulic supply systems are actually control systems with internal feedbacks and have definite dynamic properties, which in many cases have marked influence on the dynamic characteristics of the hydraulic drive. Cases of unstable hydraulic supply system operation which can degrade the hydraulic drive operating efficiency are not rare. Therefore, during hydraulic drive bench testing, it is necessary to simulate the hydraulic supply system properties which have the most significant influence on the hydraulic drives.

A simplified schematic of a typical airplane hydraulic system /39
is shown in Figure 2.3. The variable-capacity pump (or several parallel pumps) is driven from the airplane engine shaft through a speed reducer. The pump draws the working fluid from the tank, which is pressurized to 10 - 20 N/cm^2 (1 - 2 kgf/cm^2) from an air system or compressed gas bottle. Typical regulation characteristics of variable-capacity axial-plunger pumps [12] are shown in Figure 2.4.

For zero required flow rate in the hydraulic system, the supply pressure is minimal and equal to the pressurization pressure. With increase of the required flow rate, the supply pressure decreases as a result of the operation of the pump capacity regulator and the supply line hydraulic resistance, while the return pressure p_r increases because of the hydraulic resistance of the return line.

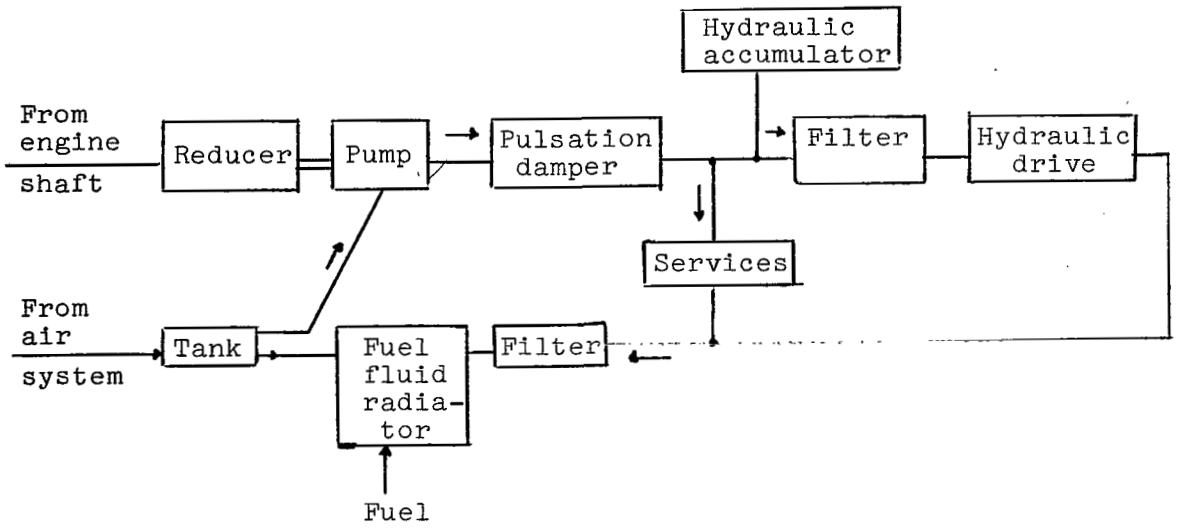

Figure 2.3. Simplified schematic of airplane hydraulic system

Thus the pressure differential across the hydraulic drive (p_s - p_r) may vary by 300 - 500 N/cm^2 (30 - 50 kgf/cm^2), depending on the required flow rate of the hydraulic drive and the other services, and in certain cases by a still larger amount.

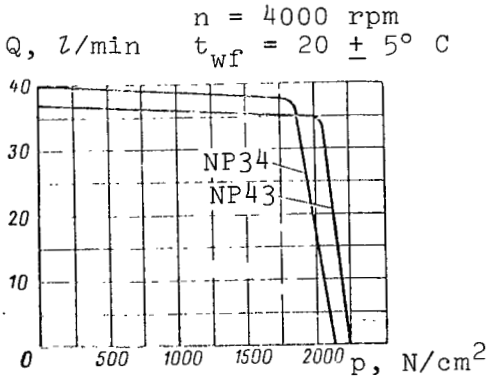

Figure 2.4. Regulation characteristics of NP34 and NP43 variable-capacity pumps

The pump provides working fluid supply with pressure pulsation, however, in aircraft hydraulic drives, the input pressure pulsations are usually small (no more than 1%) because of the use of a pulsation damper and hydraulic accumulator in the hydraulic system. /40

A pneumatic hydraulic accumulator is usually installed in the supply line in order to smooth out the pressure oscillations in case of abrupt change of the required flow rate. However, large short-term pressure overshoots with duration from several milliseconds to several seconds may occur in the lines at the hydraulic drive inlet

and outlet in spite of the presence of the hydraulic accumulator, because of the influence of variable-capacity pump inertia, line resistance, and hydraulic hammer. In this case, it is necessary to specify for the hydraulic drives, suitable requirements with respect to pressure overshoots, which may have considerable influence on hydraulic drive operating life.

The use of forced cooling of the working fluid is usually not required for the hydraulic systems of subsonic airplanes; however, a radiator which cools the working fluid by supplying fuel from the tanks is, in many cases, installed in the return line ahead of the tank for supersonic airplanes.

Working fluids of various types find application in aircraft hydraulic systems: organic, petroleum base (kerosene, AMG-10, and others), and synthetic [3, 24]. The main physical and chemical properties of working fluids of various types are presented in Table 2.1 (GOST 6794-53 and MRTU [Inter-Republic Technical Specification] 38-1-195—66). /41

We shall examine in more detail the question of hydraulic system fluid elasticity in view of its great importance for hydraulic drive dynamic characteristics. The fluid elasticity is characterized by its isothermal or adiabatic modulus of volumetric elasticity (E_i and E_a). The working fluid contains dissolved gas (primarily air) whose amount is characterized by the coefficient α, equal to the ratio of the volume of dissolved gas under normal conditions to the liquid volume. The quantities E_i and E_a are practically independent of the amount of air dissolved in the fluid, but vary somewhat as a function of working fluid temperature and pressure [23]. The pressure dependence E_a may be described by the expression: /42

$$E_a = B_a + A_a p.$$

TABLE 2.1.

Parameter	Units	AMG-10	7-50S-3
Fluid type	—	Mineral	Silicone-base synthetic
Density at pressure 10 N/cm^2 and t = 20° C	kg/cm^3	$0.85 \cdot 10^{-3}$	$0.935 \cdot 10^{-3}$
Operating temperature range (transient)	°C	From -60 to +125 (to +150)	From - 60 to +175 (to +200)
Air solubility under normal conditions	% vol	11	9.3
Kinematic viscosity at +20° C	cST	21	No less than 22
Coefficient of volumetric expansion at p = 10 N/cm^2 (1 kgf/cm^2) and 6 = 0 - 60° C	1/deg	86	78
Isothermal modulus of volumetric expansion at p = 2000 N/cm^2 (200 kgf/cm^2) and t = 20° C	N/dm^2	$1.61 \cdot 10^5$	$1.64 \cdot 10^5$
Acid number	mg KOH per g	No more than 0.05	No more than 0.1

For AMG-10 fluid at temperature 60° C, $B_a = 1.32 \cdot 10^5$ N/cm^2, and $A_a = 10.8$. In the aircraft hydraulic drive operating pressure range $(1 - 2) \cdot 10^3$ N/cm^2, the quantity $E_a = (1.42 - 1.55) \cdot 10^5$ N/cm^2.

The air solubility in the fluid is directly proportional to the pressure. Therefore, air bubbles can form in regions of reduced pressure and be entrained by the moving fluid stream as the fluid flows through the hydraulic system. The pump suction line, local resistances, and elements with unidirectional seals present the greatest danger of undissolved air release into the working fluid. The amount of undissolved air in the airplane hydraulic system fluid

depends on the hydraulic system configuration and its operating regime; however, the volumetric undissolved air content usually does not exceed 1% [2].

The presence of undissolved air may reduce noticeably the bulk elastic modulus of the fluid. However, as was shown in [23], in the pressure range typical for aircraft hydraulic drives $(1 - 2) \cdot 10^3$ N/cm^2, an undissolved air content in the fluid of less than 1% has practically no influence on the magnitude of the bulk elastic modulus of the fluid. It should be noted that even with up to 5% undissolved air content in the fluid, the adiabatic bulk modulus E_{ma} of the mixture of fluid and air decreases only slightly in this pressure range: $E_{ma} = (1.2 - 1.45) \cdot 10^5$ N/cm^2.

Thus, variation of the working fluid bulk modulus of elasticity and the air content in the fluid do not have any marked influence on the dynamic properties of aircraft hydraulic drives because of the comparatively high operating pressures and the small amount of undissolved air in the hydraulic system fluid.

The properties and state of the hydraulic supply system working fluids are very important for hydraulic drives. The question of working fluid contamination is also very important. Usually, a filter is installed in the supply line to protect the hydraulic drives. In addition, a filter is installed in the return line to protect the hydraulic system against the wear products of the hydraulic units. At the present time, metalloceramic filters are produced which filter out particles of five micron size, however; most of the serge-weave nickel-gauze mesh filters which are in use can filter out only particles larger than 10 - 12 microns. /43

Contaminant content of no more than 0.005% by weight (i.e., no more than 40 mg/liter), which corresponds to particle volumetric content to 0.01%, is permitted for working fluids in the as-delivered condition in accordance with GOST 6370-59. More severe requirements are imposed on the cleanliness of the working fluid used to fill

aircraft hydraulic systems. The working fluid at the servicing unit outlet is considered clean if there are no more than 150 particles of size from 5 to 10 microns, 25 particles from 10 to 25 microns, 10 particles from 25 to 50 microns, and 1 particle from 50 to 100 microns present per 100 cm^3 [29].

The contaminant content increases somewhat during hydraulic system operation, and then stabilizes at some level at which the contaminant concentration by weight does not exceed 0.005% [19]. The working fluid concentration is checked after every 50 hours of hydraulic system operation, and in a 100 cm^3 sample, the number of particles must not exceed 3000 with size from 5 to 10 microns, 800 from 10 to 25 microns, 150 from 25 to 50 microns, and 50 from 50 to 100 microns [22].

Thus, the working fluid of aircraft hydraulic systems contains a considerable number of particles which are commensurate in size with, or even exceed, the dimensions of the radial clearances in the slide valve pairs. Results of qualitative analysis of aircraft hydraulic system contaminant composition have shown that the contaminants consist of particles of metal, rubber, scale, lapping compounds, atmospheric dust, fibers, and so on [5, 29]. The hardness of certain contaminant compounds (quartz, for example) exceeds considerably the hardness of the materials from which the hydraulic drive slide valve pairs are manufactured. It should be noted that the existing standard for the maximum tolerable degree of working fluid contamination (0.005% by weight) is not adequately justified, particularly since it does not take into account particle size and nature, which are very important for hydraulic system serviceability. /44

2.6. Types of Tests

Bench testing, during which experimental determination of hydraulic drive parameters and quality is made with simulation of the operating conditions, is very important in the hydraulic drive development and production stages. The analysis made in the present

volume applies to all hydraulic drive inspection tests made in experimental and series production in order to monitor hydraulic drive quality (GOST 16504-70), and also research, delivery-acceptance, and other tests.

Bench tests of hydraulic drives consist of tests under normal conditions, tests of stability and resistance to external inputs (mechanical and climatic), and service life tests. All the necessary hydraulic drive parameters required by the technical documentation, including dynamic characteristics, are verified under normal conditions [i.e., with satisfaction of the Equality (2.2)]. In accordance with GOST 15150-69, normal climatic conditions are considered to be ambient air temperature 25 $\pm$ 10° C, relative humidity 45 - 80°, atmospheric pressure (8.4 - 10.6) N/cm^2. During tests under normal conditions, the working fluid temperature is usually 35 $\pm$ 10° C.

The tests under normal conditions also include checks with various (off-design) values of the factors b_{3j} and b_{4j} (for example, parameters with low or high supply and return pressures or off-design loading on the output element).

Tests for stability and resistance to external inputs are conducted to the extent determined by the nature of their influence on hydraulic drive parameters and quality indices. Tests for stability under the action of any external factor are conducted to verify normal hydraulic drive functioning and retention of its parameters during action of a factor which causes reversible parameter changes. Tests for resistance to the action of any external factor are conducted to verify normal hydraulic drive functioning and retention /45 of its parameters after the action of a factor which causes irreversible changes in the hydraulic drive functional components.

The required volume of hydraulic unit tests in the mechanical and climatic input case is presented in Chapters VI and VII.

During service life tests, we verify that the hydraulic drive remains operable while and after operating for the required service life in regimes which are as close as possible to the operating conditions. During the service life tests, mechanical inputs are not used, and of the climatic inputs, we simulate high and low working fluid and ambient temperatures.

At the present time, the sequential test technique is most widely used, in this case, the hydraulic drive being tested is subjected sequentially to all the test modes described above. The advantage of this test technique lies in the ability to determine precisely the cause of failure, if failure occurs, and in the need for simpler test stand equipment. The drawbacks of this technique are that the test conditions do not correspond to the actual operating conditions, where several external factors act simultaneously, and also the long test duration and consumption of a considerable part of the service life in the tests with climatic and mechanical inputs.

In order to accelerate the tests, we use the parallel test technique, in which checks with various input modes are made simultaneously on one of two (or several) hydraulic drives. When using this test technique, the test duration is reduced as a result of parallel conduct of studies on several hydraulic drives. The testing modes which cause practically no irreversible changes of the functional component parameters and properties or which do cause irreversible changes of the different functional component parameters or properties may be carried out on any of the hydraulic drives and in any sequence. The testing modes which lead to irreversible changes of the same functional component parameters or properties should be conducted on the same hydraulic drive.

When using the combined testing technique, the unit under test /46
is subjected simultaneously to the action of several external factors. This test technique makes it possible to reproduce more accurately the actual complex inputs to the units under test. However, for hydraulic drives, the combined tests require very complicated testing

equipment, and are rarely used. In view of the simplicity of the technical realization and the peculiarities of their influence on the hydraulic drives, it is advisable to conduct combined tests for stability under the action of high and low working fluid and ambient medium temperatures. The standard MIL-STD-810B specifies the conduct of combined "temperature plus low atmospheric pressure" and "temperature plus humidity plus low atmospheric pressure" testing for equipment with hydraulic drive in certain cases.

It is obvious that it is impossible in practice, under hydraulic bench test conditions, to simulate simultaneously the entire complex of external factors corresponding to the operating conditions. However, there is no need for this. In order to obtain, during bench tests, reliable data on the quality of hydraulic drive operation under operating conditions, it is necessary, as a result of analysis of the influence of the external factors on the hydraulic drives, to select that volume and sequence of tests which will reproduce all the factors which are hazardous from the viewpoint of the occurrence of the maximal reversible and irreversible changes of the hydraulic drive parameters or the properties of their functional components.

CHAPTER III

ANALYSIS OF HYDRAULIC DRIVE TEST STAND HYDRAULIC SUPPLY SYSTEMS

3.1. Analysis of Hydraulic Supply System with Variable Output Pump and Its Influence on Hydraulic Drive Characteristics

In view of the necessity for simulating in test stand hydraulic supply systems the most important characteristics of the airplane hydraulic systems, we need to analyze the basic static and dynamic characteristics of the hydraulic supply system with variable delivery pump. In such hydraulic supply systems, use is made of nonreversing pumps of various types with single-stage (direct action) or two-stage (servo action) output regulator, which automatically establishes the pump output equal to the required flow rate with the aid of supply pressure feedback. /47

In the pump with single-stage output regulator, the working fluid under supply pressure is fed beneath the servocylinder piston, on the other side there acts a reference mechanical or hydraulic spring. The servocylinder piston rod controls the position of the pump regulating organ (tilted plate or yoke, position of intake regulating head, and so on).

A schematic of one type of aircraft variable-delivery pump with single-stage output regulator [24] is shown in Figure 3.1. In the axial-plunger pump with tilted plate and valve-controlled distribution, the plungers have constant travel. Variable pump delivery is achieved by varying the intake cutoff position, and corresponding variation of the effective plunger travel, i.e., that part of the total travel in which working fluid is supplied to the delivery line. The cutoff position change is accomplished by displacing the regulating head, which is subject to precompression by a reference spring and the pressure supplied from the delivery line to the regulating head plunger. A restrictor is installed in the pressure feedback line to ensure stable pump operation. The basic advantages of the pump of this type are compactness, constructional simplicity, and good dynamic qualities.

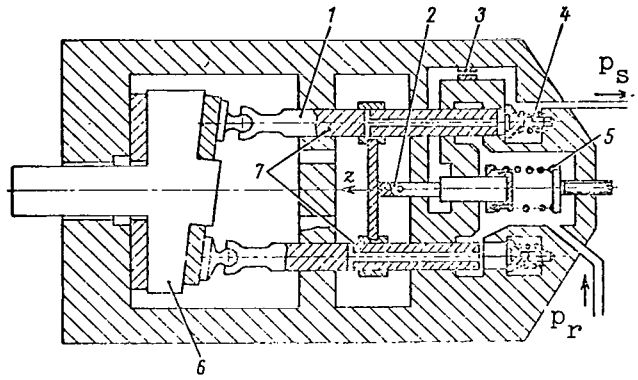

Figure 3.1. Schematic of pump with single-stage output regulator:
1 — plunger; 2 — regulating head; 3 — restrictor; 4 — valve; 5 — regulator reference spring; 6 — inclined plate; 7 — plunger bypass port

/48

A simplified schematic of a hydraulic supply system with variable-delivery pump to operate the hydraulic load is shown in Figure 3.2. Any of the hydraulic drives may serve as the hydraulic load. In dynamic analysis of the hydraulic supply system, we can take the pump drive shaft rotational speed to be constant, since the variation of this velocity is a very slow process, determined by the airplane engine operating regime. In the analysis, we neglect the influence of parameter distributedness and wave processes in the lines. For this type of pump, we can consider that the leakage in the pump is directly proportional to the supply pressure, there is practically no leakage around the regulating head plunger, and the working fluid flow in the feedback line is laminar for small pressure differentials

/49

across the restrictor. In the analysis, we neglect the forces which the plungers and working fluid stream flowing through the plunger inlet ports exert on the regulating head. We consider the pump inlet pressure to be adequate for complete filling of the working chambers below the pump plungers. As the zero position of the regulating head, we take that in which the pump output is equal to zero, and the regulator spring is compressed to the maximal degree (z_a). With these assumptions, the system of equations describing operation of the hydraulic supply system without hydraulic accumulator on the working segment of the pump regulation characteristic can be obtained in the form:

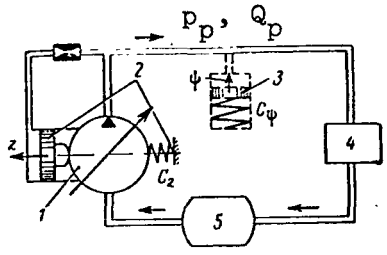

Figure 3.2. Schematic of hydraulic supply system and pump with single-stage output regulator:
1 — variable-output pump; 2 — output regulator; 3 — piston-spring hydraulic accumulator; 4 — hydraulic load; 5 — tank

$$\left.\begin{aligned} k_p z - k_{le} p_p &= Q + k_{vl} \dot{p}_p - A_z \dot{z}, \\ m\ddot{z} = F_z \dot{z} + C_z z &= C_z z_a - A_z p_p, \end{aligned}\right\} \qquad (3.1)$$

where Q is the flow rate required by the hydraulic load; $k_p = A_i\, in_i = \frac{q_0 n_i}{l_0}$ is the pump gain; A_i is the plunger section area; i is the number of plungers; n_i is the pump drive shaft rotation frequency per unit time; q_0 is the pump output per revolution; l_0 is the maximal effective plunger travel; k_{le} is the leakage coefficient; $k_{V_1} = V_1/E$ is the working fluid compressibility in the delivery line; V_1 is the volume of working fluid in the supply line; A_z is the active area of the regulating head plunger; m is the referred mass of the regulating head; C_z is the regulator spring stiffness; $F_z = f + A_z^2/G_r$ is the referred damping coefficient of the regulator; f is the regulating /50

head damping coefficient; G_r is the conductance of the pump regulator restrictor for laminar working fluid flow.

For analysis of hydraulic supply system stability, we can represent (3.1) in terms of increments relative to the steady state, characterized by the parameters Q_0, p_{p0}, z_0,

$$\left.\begin{aligned} k_p \Delta z - k_{1e} \Delta p_p &= \Delta Q + k_{V_1} \Delta \dot{p}_p - A_z \Delta \dot{z}, \\ m \Delta \ddot{z} + f_z \Delta \dot{z} + C_z \Delta z &= -A_z \Delta p_p. \end{aligned}\right\} \qquad (3.2)$$

The block diagram corresponding to these equations is shown in Figure 3.3 (solid lines). The hydraulic supply system can be represented in the form of an integrating element with direct feedback (with respect to leakage flow rate) and feedback through an oscillatory element characterizing the dynamic properties of the pump output regulator.

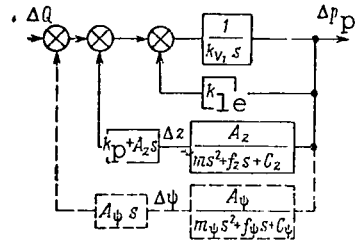

Figure 3.3. Hydraulic supply system block diagram

We can obtain the transfer function of the hydraulic supply system with variable output pump from (3.2) in the form

$$W(s) = \frac{\Delta p_p(s)}{\Delta Q(s)} = -\frac{1}{h_1} \cdot \frac{T_z^2 s^2 + T_{z1} s + 1}{A_3 s^3 + A_2 s^2 + A_1 s + 1}, \qquad (3.3)$$

where

$$T_z = \sqrt{\frac{m}{C_z}}; \quad T_{z1} = \frac{f_z}{C_z}; \quad A_3 = \frac{k_{V_1} m}{C_z h_1};$$

$$A_2 = \frac{k_{V_1} f_z + m k_{1e}}{C_z h_1}; \quad A_1 = \frac{k_{V_1} C_z + k_{1e} f_z + A_z^2}{C_z h_1};$$

$h_1 = \frac{k_p A_z}{C_z} + k_{1e}$ is the slope of the operating segment of the pump regulation characteristic. /51

Thus, the hydraulic supply system transfer function is characterized by the ratio of two second- and third-degree polynomials, and the gain is determined by the slope of the pump regulation

characteristic. If we neglect the flow used to displace the regulating head and the leakage flow, both of which promote system stability, the transfer function (3.3) takes the form

$$W(s) = -\frac{1}{h} \cdot \frac{T_z^2 s^2 + T_{z1} s + 1}{T_z^2 T_{V_1} s^3 + T_{z1} T_{V_1} s^2 + T_{V_1} s + 1}, \qquad (3.4)$$

where $T_{V_1} = \frac{k_{V_1}}{h}$ is the time constant due to the influence of working fluid compressibility in the supply line; $h = h_1 - k_{1e}$.

The hydraulic supply system frequency characteristics are shown in Figure 3.4 for different values of T_z, T_{z1}, T_{V_1}. For an ideal output regulator ($T_z = T_{z1} = 0$), the pump is described by an inertial element with time constant T_{V_1} (curves 1). In the weakly damped pump regulator case (curves 2), there is a resonance peak at a frequency close to the regulator natural frequency $\omega = 1/T_z$. With increase of regulator damping, the resonance peak decreases and shifts toward the lower frequencies (curves 3). In the overdamped regulator case (curves 4), there is a low-frequency resonance peak and the resonance peak frequency:

$$f_r \approx \frac{1}{2\pi} \sqrt{\frac{C_z h}{k_{V_1} f_z}}.$$

If the working fluid volume in the supply line is increased in the case with weak pump regulator damping (curves 5), the pump passband narrows, the resonance peak disappears, and the pump dynamic characteristics approach those of the pump with ideal regulator.

Analyzing hydraulic supply system stability with the aid of the Hurwitz criterion, we can obtain the following stability condition: /52

$$A_1 A_2 > A_3$$

or

$$m < \frac{k_{V_1} f_z (k_{V_1} C_z + k_{1e} f_z^2 + A_z^2)}{k_{V_1} k_p A_z - k_{1e} A_z^2 - k_{1e}^2 f_z} \tag{3.5}$$

With decrease of the leakage and regulating head displacement flow, the stability region narrows, and approaches the form:

$$f_z > \frac{mh}{k_{V_1}} \tag{3.6}$$

or

$$T_{z_1} > \frac{T_z^2}{T_{V_1}}. \tag{3.7}$$

To each combination of pump design parameters there corresponds a critical value of regulator damping, upon exceeding which the hydraulic supply system will be stable. The critical damping increases with increase of the regulating head mass and the pump regulation characteristic slope, and also with decrease of the working fluid volume in the delivery line. With pump regulator damping less than the critical value, undamped oscillations develop in the hydraulic supply system, and their amplitude is limited by regulating head travel. Since the hydraulic supply source pressure oscillations

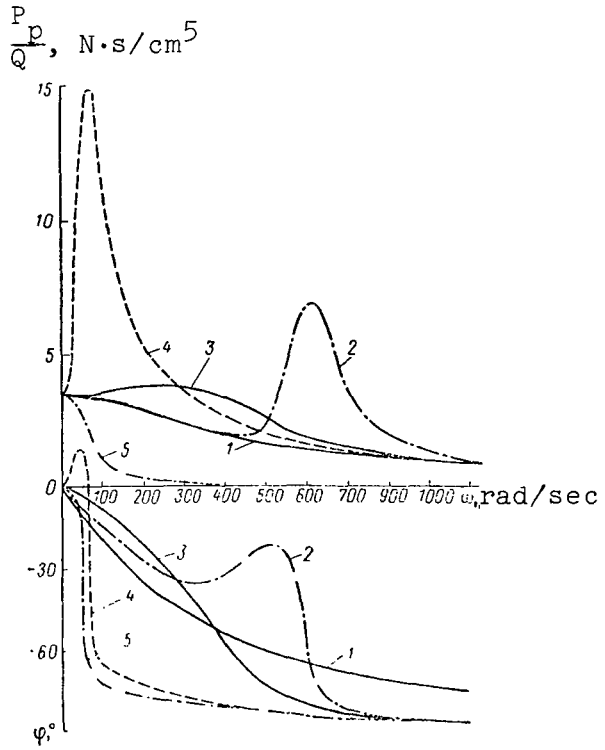

Figure 3.4. Hydraulic supply system frequency characteristics

due to its instability are usually high-frequency (30 - 50 Hz), these oscillations will not have any marked influence on hydraulic drive output member displacement. The danger of these pressure oscillations lies in reduction of system reliability, since all the components operate in the vibrational regime, resonant phenomena in the system are possible, and heating and "churning" of the working fluid increases. In order to avoid self-maintained oscillations in the hydraulic supply system in any operating regime, and in order that the hydraulic supply system have minimal influence on hydraulic drive dynamic characteristics, we should select the output regulator damping magnitude [27] /53

$$f_z = (4 - 6)\frac{mh}{k_{V_1}}$$

or the restrictor conductance

$$G_r = (0.15 - 0.25)\frac{A_z^2 V_1}{mhE}.$$

Let us examine the influence on the dynamic properties of a hydraulic supply system with variable-output pump of a hydraulic accumulator installed in the supply line. As an example for analysis, /54 we consider a piston-spring hydraulic accumulator, shown schematically in Figure 3.2. The equation of the forces acting on the hydraulic accumulator piston has the form

$$m_\psi \ddot{\psi} + f_\psi \dot{\psi} + C_\psi \psi - C_\psi \psi_a = A_\psi p_p - F_\psi \operatorname{sing} \dot{\psi},$$

where ψ is the hydraulic accumulator piston displacement; m_ψ, f_ψ are the piston mass and damping; A_ψ is the piston area; C_ψ is the hydraulic accumulator spring stiffness; ψ_a is the spring compression at the zero piston position; F_ψ is the piston friction force.

For this case, the linearized equations of the hydraulic supply system will be (for $F_\psi = 0$):

$$\left.\begin{aligned} k_p \Delta z - k_{1e} \Delta p_p &= \Delta \dot{Q} - A_z \Delta \dot{z} + k_{V_1} \Delta \dot{p}_p - A_\psi \Delta \dot{\psi}, \\ m \Delta \ddot{z} + f_z \Delta \dot{z} + C_z \Delta z &= -A_z \Delta p_p, \\ m_\psi \Delta \ddot{\psi} + f_\psi \Delta \dot{\psi} + C_\psi \Delta \psi &= -A_\psi \Delta p_p. \end{aligned}\right\} \qquad (3.8)$$

As a consequence of the influence of the hydraulic accumulator, rate feedback through an oscillatory element characterizing the inertia of the hydraulic accumulator piston (shown in Figure 3.3), is added to the hydraulic supply system block diagram.

If the hydraulic accumulator is considered ideal (i.e., if $m_\psi = f_\psi = 0$), then in the first equation of (3.8) we can replace the term $(-A_\psi \Delta \dot{\psi})$ by the expression $k_\psi \Delta \dot{p}_p$, where $k_\psi = A_\psi^2/C_\psi$. The influence of the coefficient k_ψ on hydraulic supply system dynamic properties is analogous to the influence of working fluid compressibility in the supply line. However, since in the presence of the hydraulic accumulator $k_\psi >> k_{V_1}$, the influence of the coefficient k_{V_1} can be neglected. Then the hydraulic supply system stability condition takes the form

$$f_z > \frac{mh}{k_\psi}.$$

It can be shown that hydraulic accumulator piston inertia and damping broaden somewhat the region of hydraulic supply system reliability. In the presence of a sufficiently large hydraulic accumulator ($k_\psi > 0.1 — 0.5$ cm^5/kgf), dynamic supply pressure changes resulting from the pump output regulator influence are practically zero for any output regulator design parameters. /55

It should be noted that the analysis made also applies to the pneumatic hydraulic accumulator, since it is similar in its dynamic characteristics to the ideal piston hydraulic accumulator.

In all pumps with single-stage output regulator under steady state conditions, the magnitude of the supply pressure varies as a function of the required flow rate. If it is necessary to maintain a constant pressure, pumps with two-stage pressure regulator containing an integrating element are used [3, 24]. It should be noted that in practice, even for a pump with two-stage regulator, the magnitude of the supply pressure changes somewhat depending on the flow rate required, and the slope of the pump regulation characteristic is determined by the slope of the regulator control slide valve force characteristic and the rate of change of the hydraulic force on the pump regulating organ.

As a result of analysis of the dynamic properties of the hydraulic supply system with a pump having a two-stage output regulator, we find that they are analogous to the previously examined dynamic properties of the hydraulic supply system with a pump having a single-stage output regulator (since the influence of inertia of the regulator first amplification stage in the working frequency range is small because of the small mass of the first stage regulating element).

Now we shall examine the influence of the hydraulic supply system on hydraulic drive static and dynamic characteristics. As an example, we consider the influence of a pump with single-stage regulator on the characteristics of the slide valve actuator with symmetric piston [27]. For this case, we can describe the hydraulic drive, with account for the hydraulic supply system influence, using (1.1) and (3.1), by the differential equations: /56

$$\left.\begin{aligned} &k_3 x\sqrt{p_p - p_R \operatorname{sing} x} = A_p\dot{y} + k_V\dot{p}_R,\\ &A_p p_R = M\ddot{y} + f_y\dot{y} + C_p y + R,\\ &k_p z = k_3|x|\sqrt{p_p - p_R \operatorname{sing} x} + k_{V_1}\dot{p}_p + \\ &\qquad + k_{1e}p_p - A_z\dot{z},\\ &m\ddot{z} + f_z\dot{z} + C_z z = Cz_a - A_z p_p. \end{aligned}\right\} \qquad (3.9)$$

Solving this system of equations for the steady state motion regimes, we can obtain the expression for the actuator mechanical characteristic

$$v = \frac{k_3^2}{2A_p h} x^2 \left(\sqrt{1 + \frac{4h^2 (p_m - p_R \operatorname{sign} x)}{k_3^2 x^2}} - 1 \right) \operatorname{sing} x, \qquad (3.10)$$

where $p_m = \frac{C_z z_a}{A_z}$ is the zero-output pressure.

In order to evaluate the influence of pump regulation characteristic slope on the actuator static characterics, it is convenient to introduce the parameter κ, equal to the ratio of the delivery pressure at the maximal required flow rate to the maximal delivery pressure in the system. Converting to relative quantities, we can obtain from (3.10) the mechanical characteristic

$$\bar{v} = \frac{1-\varkappa}{2\varkappa} \bar{x}^2 \left(\sqrt{1 + \frac{4\varkappa (1 - \bar{p}_R \operatorname{sign} \bar{x})}{(1-\varkappa)^2 \bar{x}^2}} - 1 \right) \operatorname{sing} \bar{x}, \qquad (3.11)$$

where $\bar{v} = \frac{v}{v_m}$; $\bar{p}_R = \frac{p_R}{p_m}$; $\bar{x} = \frac{x}{x_m}$.

This characteristic is shown for various κ's in Figure 3.5. With reduction of the pump regulation characteristic slope, the actuator mechanical characteristic stiffness decreases somewhat. Taking $\bar{p}_R$ = 0 in (3.11), we obtain the actuator velocity characteristic shown in Figure 3.6. With reduction of the pump regulation characteristic slope, the actuator piston velocity (for the same maximal velocities) increases, the nonlinearity of the velocity characteristic increases, the gain $\bar{k}_{act} = \frac{d\bar{v}}{d\bar{x}}$ increases for small

/57

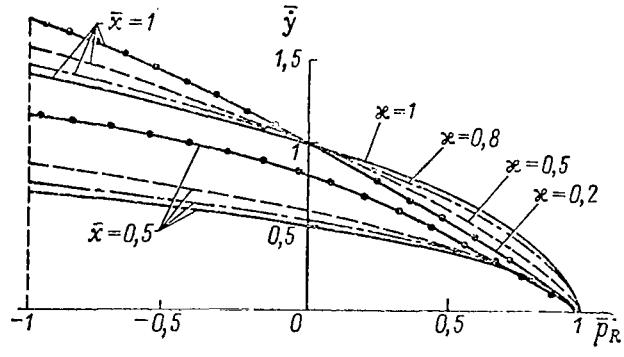

Figure 3.5. Actuator mechanical characteristic with account for pump regulation characteristic slope

signals and decreases for large signals, and the range of $\bar{k}_{act}$ variation increases with decrease of κ.

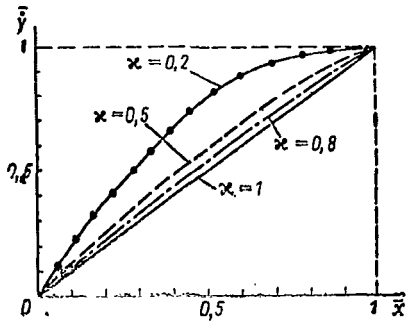

Figure 3.6. Actuator velocity characteristic with account for pump regulation characteristic slope

We can also find from (3.9) that the delivery pressure in the system depends linearly on the actuator piston velocity

$$p_p = p_m - \frac{A_p}{h}|v|.$$

Let us examine the hydraulic supply source influence on the frequency characteristics and transient processes in the hydraulic drive described by (3.9). The block diagram corresponding to this system of equations is shown in Figure 3.7. The salient feature of this system is the presence of the characteristic modulus-type nonlinearity in the delivery line flow balance equation (or in the feedback with respect to the hydraulic supply source in the block diagram). /58 Physically, the modulus sign indicates that the pump flow rate is independent of the actuator piston motion direction, i.e., regardless of which of the two slide valve ports the working fluid enters from the delivery line.

Analyzing the stability of monotonic actuator piston motion (i.e., motion with sign-constant piston velocity), we can obtain from (3.9) the linearized equations:

$$\begin{cases} k_x \Delta x + k_p \Delta p_p - k_p \Delta p_R = A_p \Delta \dot{y} + k_V \Delta \dot{p}_R, \\ A_p \Delta p_R = M \Delta \ddot{y} + f_y \Delta \dot{y} + C_p \Delta y, \\ k_p \Delta z - k_{le} \Delta p_p = k_x \Delta x + k_p \Delta p_p - k_p \Delta p_R + \\ \qquad + k_{V_1} \Delta \dot{p}_p - A_z \Delta \dot{z}, \\ m \Delta \ddot{z} + f_z \Delta \dot{z} + C_z \Delta z = - A_z \Delta p_p. \end{cases}$$

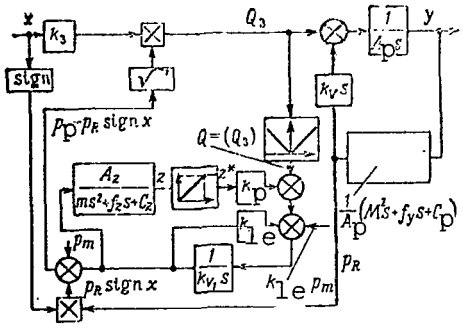

Figure 3.7. Block diagram of actuator with account for influence of hydraulic supply system with variable output pump

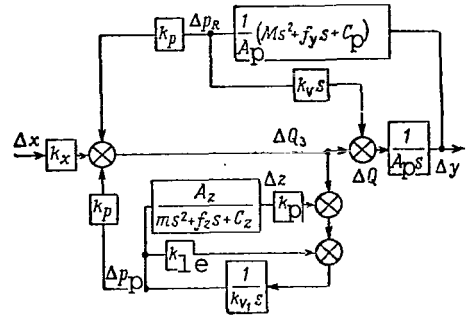

Figure 3.8. Block diagram of linearized actuator model with account for hydraulic supply system influence

As a consequence of the hydraulic supply source influence, in the linearized system block diagram shown in Figure 3.8, feedback is added with respect to the hydraulic supply source which contains the element described by (3.3). Convolving the loop with this feedback, /59 we find that, as a consequence of the hydraulic supply system influence, there is added in the linearized actuator block diagram forward circuit an element with transfer function equal to the ratio of two third-degree polynomials. The presence of this element causes sequential peaks and valleys on the actuator AFC, and additional phase lag at frequencies close to resonance on the PFC, where the intensity of the amplitude and phase distortions introduced by the hy- /60 draulic supply system decreases with decrease of the control signal magnitude and increase of the delivery line elasticity.

As a result of analysis of the stability of this system with the aid of the Hurwitz criterion, we find that the minimal stability region occurs for $f_y = 0$ and $x_0 \to 0$, and this stability region coincides with the stability region of the hydraulic supply source itself, described by (2.10) and (2.11).

Let us examine in more detail the influence of modulus-type nonlinearity on the system dynamic characteristics. We shall analyze forced oscillations in the unloaded hydraulic drive with sinusoidal

input signal. For an ideal hydraulic supply source ($T_z = T_{zl} = T_{V_1} = 0$), the equations of flow rate $\bar{Q}$ and delivery pressure $\bar{p}_p$, expressed in relative units, can be obtained in the form: /61

$$\bar{Q} = \frac{1-\varkappa}{2\varkappa}\bar{x}_0^2\left(\sqrt{1+\frac{4\varkappa}{(1-\varkappa)^2\bar{x}_0^2\sin^2\omega t}}-1\right)\times \\ \times \sin^2\omega t \operatorname{sing}\sin\omega t, \tag{3.12}$$

$$\bar{p}_p = 1-\frac{(1-\varkappa)^2}{2\varkappa}\bar{x}_0^2\left(\sqrt{1+\frac{4\varkappa}{(1-\varkappa)^2\bar{x}_0^2\sin^2\omega t}}-1\right)\times \\ \times \sin^2\omega t, \tag{3.13}$$

where $\bar{x}_0$ is the input signal amplitude.

We see from (3.12) and (3.13) that the forced oscillations of the delivery pressure and pump regulating element displacement have a dual frequency and are close in form to the rectified sinusoid (3.9). With decrease of the pump regulation characteristic slope, the delivery pressure oscillation amplitude increases, and the oscillations of $\bar{v}$ (or of $\bar{Q}$), which are nearly sinusoidal, alter in form, so that the oscillation half-waves become fuller, approaching rectangular waves, while the period of the $\bar{v}$ oscillations does not change.

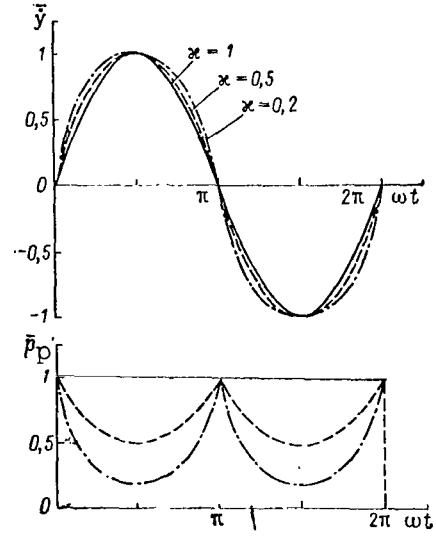

Figure 3.9. Forced actuator oscillations as function of velocity and hydraulic supply system pressure

We see from the block diagram shown in Figure 3.7 that the hydraulic supply source input signal is the absolute value of the working fluid flow rate through the slide valve $Q^* = |Q|$. In the case of sinusoidal actuator flow rate oscillations, the change Q^* can be found by expanding it into a Fourier series [32]:

$$Q^*(t) = Q_0 |\sin \omega t| = a_0 + \sum_{n=1}^{\infty} (a_n \cos n\omega t + b_n \sin n\omega t),$$

where

$$a_0 = \frac{Q_0}{\pi} \int_0^{\pi} |\sin \omega t| \, d\omega t;$$

$$a_n = \frac{2Q_0}{\pi} \int_0^{\pi} |\sin \omega t| \cos n\omega t \, d\omega t;$$

$$b_n = \frac{2Q_0}{\pi} \int_0^{\pi} |\sin \omega t| \sin n\omega t \, d\omega t.$$

In the Fourier series for the function Q* (t), terms with sines and all the odd harmonics are missing. The function Q* (t) can be represented in the form of a constant component and even harmonics: /62

$$Q^*(t) = \frac{2Q_0}{\pi} - \frac{4Q_0}{3\pi} \cos 2\omega t - \frac{4Q_0}{15\pi} \cos 4\omega t -$$
$$- \frac{4Q_0}{35\pi} \cos 6\omega t - \ldots = 0{,}637 Q_0 - 0{,}425 Q_0 \cos 2\omega t -$$
$$- 0{,}085 Q_0 \cos 4\omega t - 0{,}036 Q_0 \cos 6\omega t - \ldots.$$

Thus, in the case of sinusoidal forced hydraulic drive oscillations, there are oscillations of the delivery pressure and pump regulating element with frequencies corresponding to even harmonics of the forced oscillations. The form and amplitude of these oscillations depend on the hydraulic supply system frequency characteristic. Usually, the hydraulic supply system is a low-frequency filter; therefore, with increase of the forced oscillation frequency, the delivery pressure oscillation amplitude decreases, with the primary reduction being that of the degree of influence of the higher even harmonics. For approximate analysis of hydraulic supply system influence on actuator forced oscillations, we can limit ourselves to only the first two terms of the expansion of the function Q* into a Fourier series. In this case, with sinusoidal forced oscillations of the hydraulic drive delivery pressure and regulating element oscillations occur with twice the frequency. Depending on the hydraulic supply system frequency characteristics, the magnitude and phase of the pressure oscillations will be different, and, consequently, they will have a different effect on the output signal.

The equation of working fluid flow rate in the unloaded actuator can be obtained in the form:

$$Q \approx k_3 x_0 \times \times \sqrt{p_m - \frac{0{,}637 Q_0}{h} + \frac{0{,}425 Q_0}{h} A(\omega) \cos[2\omega t - \varphi(\omega)]} \times \times \sin \omega t, \qquad (3.14)$$

where A (ω) and ϕ (ω) are determined from the hydraulic supply system frequency characteristic.

For the hydraulic supply system transfer function defined by (3.5), we can obtain /63

$$A(\omega) = \sqrt{\frac{(1 - 4T_z^2 \omega^2)^2 + 4T_{z1}^2 \omega^2}{(1 - 4A_2 \omega^2)^2 + (2A_1 \omega - 8A_3 \omega^3)^2}};$$

$$\varphi(\omega) = \operatorname{arctg} \frac{2T_{z1}\omega}{1 - 4T_z^2 \omega^2} - \operatorname{arctg} \frac{2A_1 \omega - 8A_3 \omega^3}{1 - 4A_2 \omega^2}.$$

Equation (3.14) can be solved by successive approximations.

For a weakly damped pump regulator in the range of frequencies exceeding by a factor of two the regulator natural frequency, the delivery pressure oscillation amplitude increases ($A(\omega) \gg 1$), and there is reduction of the oscillation amplitude on the hydraulic drive AFC. A similar pattern holds for an overdamped pump regulator, but, in this case, the pressure oscillation amplitude increase and the corresponding actuator flow rate oscillation amplitude reduction take place in the low-frequency region. When the control signal frequency exceeds considerably the hydraulic supply system cutoff frequency, the harmonic components of Q* are not passed by the hydraulic supply system, and only the constant component is passed. This this case, a constant delivery pressure is established in the system, equal to

$$p_{d0} = p_m - \frac{0{,}2 k_3^2 x_0^2}{h^2} \left(\sqrt{1 + \frac{9{,}8 h^2 p_m}{k_3^2 x_0^2}} - 1 \right), \qquad (3.15)$$

and the flow rate oscillations have a sinusoidal form with oscillation amplitude:

$$Q_0 = \frac{0{,}32 k_3^2 x_0^2}{h}\left(\sqrt{1 + \frac{9{,}8 h^2 p_m}{k_3^2 x_0^2}} - 1\right).$$

It can be shown that the oscillation amplitude Q_0 in the latter case exceeds somewhat the oscillation amplitude Q for the ideal hydraulic supply system with variable-output pump. If there is a hydraulic accumulator in the system, the hydraulic supply source pass-band narrows considerably. As a result of this, the hydraulic supply system does not pass the harmonic components of the oscillations in the hydraulic drive operating frequency zone, and a constant delivery pressure p_{d0}, corresponding to (3.15), is established in the system. /64

For oscillation frequencies at which the amplitude and phase distortions of the delivery pressure curve are large, the shape of the actuator flow rate variation curve may change somewhat. Since the pressure oscillations take place at twice the frequency and the distortions of each sinusoidal input signal half-wave are the same, and the maximal deviations of the output signal coincide on both half-waves, the null value of the output signal coincides in time with the null value of the input signal and is independent of the hydraulic system parameters. As a consequence of this, in the case being considered the phase shift between the input and output signals is equal to zero for any hydraulic supply source parameters, while the output signal oscillation amplitude and shape change somewhat, depending on the hydraulic supply system parameters. At frequencies corresponding to small phase shifts and the peak on the hydraulic supply system AFC, the actuator velocity oscillation amplitude decreases because of the hydraulic supply system influence. The degree of this reduction is larger, the larger the ratio of the pressure oscillation amplitude to the delivery pressure amplitude in the system with stationary actuator piston. At frequencies corresponding to phase shifts in the hydraulic supply system close to 90° for large delivery pressure oscillation amplitudes, the output signal shape is distorted so that the output signal maximum is shifted somewhat in each half-period.

A more detailed analysis of the frequency characteristics and transient processes in an inertially loaded hydraulic drive was carried out using the MN-7 simulator. In the analysis, it was assumed that when the pump regulating head reaches the stop, there is absolutely inelastic impact, during which the regulating head stops instantaneously upon reaching the stop, and then motion of the regulating head again starts with zero initial conditions.

In realizing the model corresponding to (3.9), we used standard resolving amplifiers, multiplication block, square root extraction block, modulus block, and high-sensitivity two-position polarized relays which react to the signs of the slide valve and regulating head displacements. /65

We first examined the unloaded hydraulic drive frequency characteristics for different hydraulic supply system parameters. The data obtained as a result of modeling hydraulic supply system parameter influence on the form of the forced oscillations with respect to output member velocity, delivery pressure, and regulating head displacement confirm the validity of the previously conducted approximate analysis. If there are self-maintained oscillations in the hydraulic supply system (system is unstable), on the basic hydraulic drive piston velocity oscillation component there is superposed a high-frequency component whose frequency is equal to the auto-oscillation frequency, and whose amplitude varies from zero for zero control signal to the maximal value for the maximal signal.

In all the cases examined, there was no phase shift between input signal and output member velocity. This is the fundamental difference between this nonlinear system and the previously examined linearized system, in which there was a phase shift due to the presence of the hydraulic supply system. The hydraulic supply system influences hydraulic drive AFC through the delivery pressure oscillations, which have twice the driving frequency. In the region of frequencies half the hydraulic supply system resonant frequency, there appears a dip on the hydraulic drive AFC, while at high

frequencies the AFC is somewhat higher because of the fact that a constant delivery pressure corresponding to the Q* zero harmonic is established in the system. Typical AFC of hydraulic drives with different hydraulic supply system parameters are shown in Figure 3.10.

Thus, as a result of the presence of a nonlinear modulus-type function, the hydraulic supply system influence on hydraulic drive frequency characteristics decreases in comparison with the results of linearized model analysis, and the hydraulic supply system has the greatest influence on the hydraulic drive AFC not at the resonant frequency, but at a frequency lower by a factor of two. The nature of hydraulic supply system influence on forced oscillations in the loaded hydraulic drive is similar.

As a result of simulator analysis of the transient processes /66
with respect to hydraulic drive velocity, it was found that the hydraulic supply system may have a marked influence on the transient processes with respect to hydraulic drive velocity, and the degree of this influence is larger, the larger the control signal magnitude. Damped or undamped oscillatory components, respectively, appear in the transient process for the weakly damped and/or unstable hydraulic supply system. In the case of excessive pump regulator damping, there appears in the initial segment of the transient process a dip in the delivery pressure and the transient process, with respect to piston velocity, is stretched out somewhat in comparison with the transient process for an ideal hydraulic supply system. The hydraulic supply system has a similar influence on the transient processes with respect to velocity in the loaded hydraulic drive. The results of experiments conducted on a hydraulic supply system without accumulator and with variable-output pump and a single-stage pressure regulator confirm the relations presented above. We see clearly on the oscillograms shown in Figure 3.11, the doubling of the z and p_d oscillation frequency, and the smoothing of the shape of these curves at the 10 Hz frequency. With frequency increase, the phase lag be- /68
tween the delivery pressure curve and the control current curve increases which, however, has practically no influence on the phase

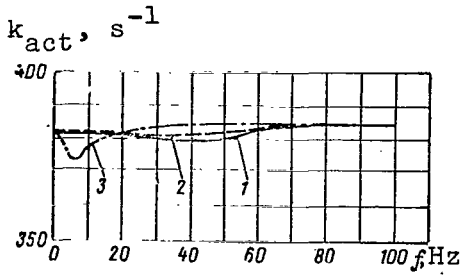

Figure 3.10. Amplitude frequency characteristics of unloaded actuator with account for hydraulic supply system influence:

1 — for weakly damped pump output regulator; 2 — for nearly optimal damping; 3 — for regulator overdamping

shift between the control current and the hydraulic drive piston output displacement. The oscillogram of the transient process in the hydraulic drive with unstable hydraulic supply system is shown in Figure 3.12. We see from this oscillogram that high-frequency pressure oscillations in the hydraulic supply system have practically no influence on hydraulic drive output member motion. On the basis of the relations presented in this study, hydraulic supply system stability was ensured by increasing pump regulator damping.

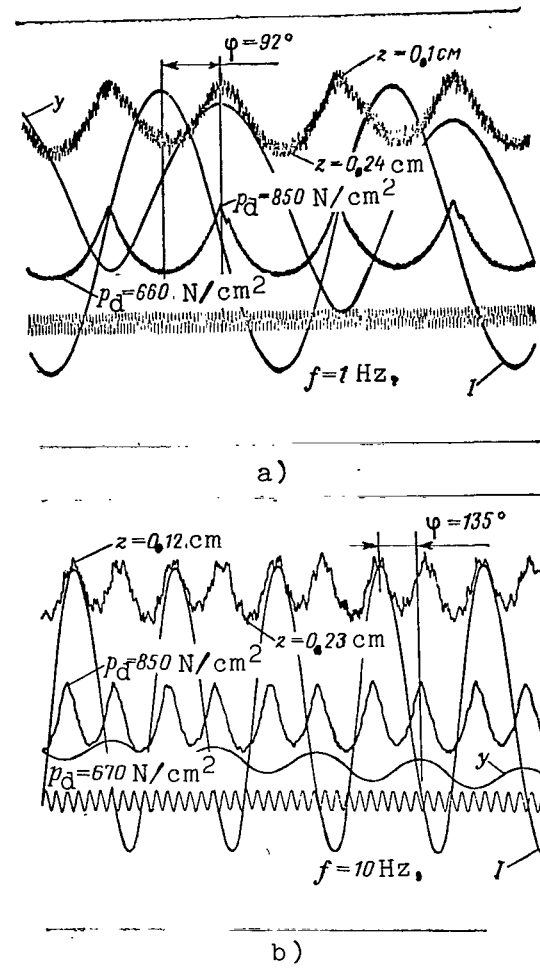

Figure 3.11. Experimental oscillogram of forced oscillations of servocontroller operating from hydraulic supply system with variable-output pump:

a — for f = 1 Hz; b — for f = 10 Hz

We shall now use the phase-plane method to analyze for some particular examples, the influence of a hydraulic supply system with variable output pump on the transient processes in a booster and electrohydraulic control surface drive. For a hydraulic supply system without accumulator, small regulating element mass, and incompressible working fluid, the system of equations describing booster motion with inertial load has the form:

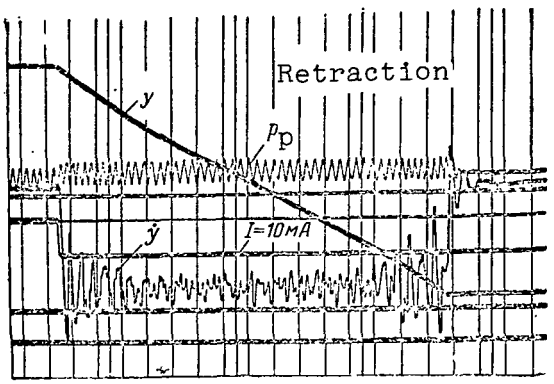

Figure 3.12. Experimental oscillogram of transient process in servocontroller operating from unstable hydraulic supply system

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$$\begin{cases} k_3 x \sqrt{p_p - \frac{M}{A_p}\ddot{y}\,\mathrm{sign}\,x} = A_p\dot{y}, \\ k_{in}\, y_{in} - k_{fb}\, y = x, \\ k_p\, z = A_p\,|\dot{y}|, \\ f_z\dot{z} + Cz = Cz_a - A_z p_p. \end{cases}$$

For analysis of booster free motions on the phase-plane, this system of equations can be reduced to the form:

$$\bar{v} = -\bar{y}\sqrt{1 - a_1\bar{v}\,\mathrm{sign}\,\bar{v} - a_2\bar{v}\,\mathrm{sign}\,\bar{v} - a_3\dot{\bar{v}}\,\mathrm{sign}\,\bar{v}}, \qquad (3.16)$$

where $\bar{y} = \frac{y}{y_0}$; $\bar{v} = \frac{v}{y_0 D_m}$; y_0 is the initial piston rod displacement;

$D_m = \frac{k_3\, k_{fb}\, \sqrt{p_m}}{A_p}$ is the booster figure of merit for maximal pressure;

$a_1 = \frac{A_p\, y_0\, D_m}{p_m\, h}$ is a coefficient characterizing the pump regulation characteristic slope influence;

$a_2 = \frac{M\, y_0\, D_m}{A_p\, p_m}$ is a coefficient characterizing the inertial load influence;

$a_3 = \frac{A_p\, f_z\, y_0\, D_m}{k_{pu} A_z\, p_m}$ is a coefficient characterizing the pump regulator damping influence.

Considering that $\ddot{\bar{v}} = \frac{d\dot{\bar{v}}}{dt} = \frac{d\dot{\bar{v}}}{d\bar{y}} \cdot \frac{d\bar{y}}{dt} = \frac{d\dot{\bar{v}}}{d\bar{y}} \bar{v} D_m$, we can reduce (3.16) to the expression:

$$\dot{\bar{v}} = -\bar{y} \sqrt{1 - \left[a_1 + D_m (a_2 + a_3) \frac{d\bar{v}}{d\bar{y}}\right] |\bar{v}|}\,.$$

The phase portrait of such a system can be plotted by the isocline method, taking $d\bar{v}/d\bar{y} = N$; then the isocline equation takes the form: /70

$$\bar{y} = -\frac{\bar{v}}{1 - [a_1 + D_m N (a_2 + a_3)]\,|\bar{v}|}.$$

By taking various values of N, we can plot the family of isoclines, and from them determine the family of phase trajectories: Line 1 in Figure 3.13 shows the phase trajectory for an unloaded booster supplied from a constant pressure pump ($a_1 = a_2 = a_3 = 0$).

The equation of this phase trajectory is: $\bar{v} = \bar{y}$. In this case, the transient process is an exponential

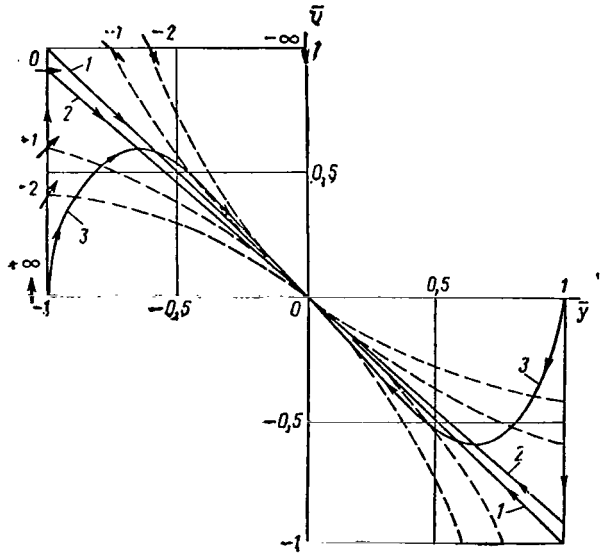

Figure 3.13. Phase portrait of booster with account for hydraulic supply system influence

$y = y_0 e^{-\frac{t}{T}}$, with the time constant $T = \frac{1}{D_m} = \frac{A_p}{k_3 \sqrt{p_m} k_{fb}}$.

Curve 2 shows the phase trajectory for a sloping pump regulation characteristic. The phase trajectory lies closer to the $\bar{y}$ axis than the phase trajectory for constant delivery pressure. This indicates that the transient process for the sloping pump characteristic lasts longer. Figure 3.13 also shows the family of isoclines (shown dashed) and the phase trajectories (curves 3) with account for regulator damping and inertial load. We see from phase plane analysis that, as a consequence of the influence of M and F_z, the booster output member velocity decreases in the acceleration segment, and increases somewhat in the deceleration segment. We note that the influence of M and f_z is simlar for an incompressible working fluid. /71

It is also of interest to analyze the influence of pump regulation characteristic slope on the transient processes in an unloaded control surface actuator drive with electron-hydraulic amplifier, described by the inertial element:

$$\left.\begin{aligned} k_3 x \sqrt{p_p} &= A_p \dot{y}, \\ k_p z &= A_p |\dot{y}|, \\ Cz &= Cz_a - A_z p_p, \\ T_{eha} \dot{x} + x &= k_{eha} u, \\ u &= u_{in} - k_{fd} y, \end{aligned}\right\} \qquad (3.17)$$

where u_{in} is the amplifier input voltage; k_{eha}, T_{eha} are the gain and time constant of the electron-hydraulic amplifier.

For analysis on the phase plane, this system of equations may be reduced to the form:

$$T_{eha} D_m \frac{d\bar{v}}{d\bar{y}} \bar{v} = \left[-\bar{v} - \bar{y} \sqrt{1 - a_1 |\bar{v}|} \right] \frac{1 - a_1 |\bar{v}|}{1 - 0{,}5 a_1 |\bar{v}|}, \qquad (3.18)$$

where $D_m = \frac{k_{eha} k_3 \sqrt{p_m} k_{fb}}{A_p}$

Figure 3.14 shows the isocline families and phase portraits of the system for constant supply pressure ($a_1 = 0$) and for a sloping pump regulation characteristic, obtained from (3.18). In both cases, there is a stable focus at the coordinate origin, which indicates the presence of oscillatory transient processes in the system. For the sloping pump regulation characteristic, the isoclines lie closer to the $\bar{y}$ axis than the corresponding isoclines for p_p = const. Because of this, the piston velocity for any piston positon with $p_p = p_m$ = const is higher than for the sloping pump regulation characteristic. The switching line on the phase portrait of the control surface actuator piston motion for the sloping pump regulation characteristic is the $\bar{y}$ axis. However, since for any h, the phase trajectories cross the $\bar{y}$ axis perpendicularly, the presence of modulus-type nonlinearity in this system does not cause a break in the phase trajectories when passing through this nonlinearity. /72

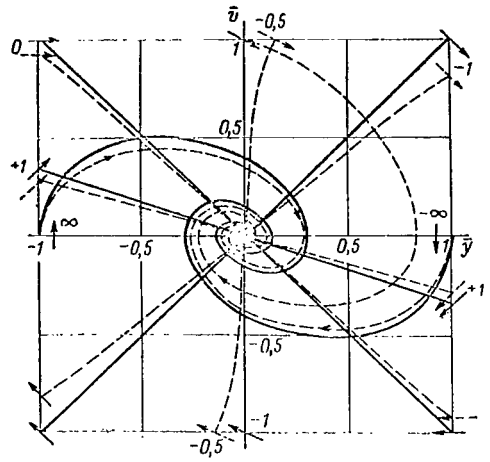

Figure 3.14. Phase portrait of unloaded electrohydraulic control surface drive:
——— — for ideal hydraulic supply system (p_p = const);
– – – — for sloping pump regulation characteristic

As a result of analysis of the transient processes corresponding to these phase trajectories and constructed graphoanalytically by approximating the phase trajectories by rectilinear segments, we find that with pump regulation characteristic slope reduction, the transient process damping increases while the oscillation frequency decreases somewhat. /73

The dynamics of the electrohydraulic control surface drive described with account for hydraulic supply system influence by (3.9) and the last two equations of (3.17) were also studied on the MN-7

simulator. As a result of the analysis, we found that the hydraulic supply system influence on the transient processes, with respect to control surface drive output member position, is negligibly small. Even for an unstable hydraulic supply source as a consequence of the filtering properties of the power cylinder, which is an integrating element, the high-frequency pressure oscillations have practically no influence on output member displacement.

If there is a hydraulic accumulator in the hydraulic supply system, or if the delivery line is sufficiently elastic, delivery pressure oscillations are practically absent in the case of force oscillations. In this case, the variable output pump influence shows up in reduction of the delivery pressure and hydraulic drive gain and, consequently, reduction of the control surface actuator figure of merit with increase of the input signal, which leads to increase of drive damping and dynamic error.

3.2. Influence of Bypass Valve in Hydraulic Supply System with Constant Output Pump on Hydraulic Drive Characteristics

In spite of the fact that airplane hydraulic drives operate with variable supply and return pressures, the requirements on permissible hydraulic drive parameter magnitudes are established for constant values of the supply and return pressures. This leads to the necessity for using test stand hydraulic supply systems which can maintain the required constant supply and return pressure values. Usually, this is achieved by using a hydraulic system with constant output pump and a bypass valve. A simplified schematic of such a /74
hydraulic supply system is shown in Figure 3.15. The pump output exceeds the maximal required hydraulic drive flow rate. The excess fluid flow rate supplied by the pump is constantly bypassed to the tank by a slide-type bypass valve. Such a hydraulic supply system is less economical than the system with variable output pump, and requires a powerful working fluid cooling system. However, in spite of these drawbacks, the hydraulic supply system with constant output

pump is the basic hydraulic supply system type for test stands used for determining hydraulic drive controllable parameters.

Let us examine the operation of a slide-type direct-action bypass valve. The supply pressure acts on one end of the bypass valve slide and the force of a spring, whose precompression determines the nominal supply pressure to be maintained, acts on the other end. The valve slide establishes itself in a position in which the magnitude of the slide slot provides, at the operating pressure, bypass into the return line of that part of the fluid flow rate which is not required by the hydraulic load. With increase of the required flow rate, the supply pressure decreases and the valve slide shifts under the action of the apring, blocking the slide slot, and thus the flow rate through the valve decreases and the supply pressure is maintained.

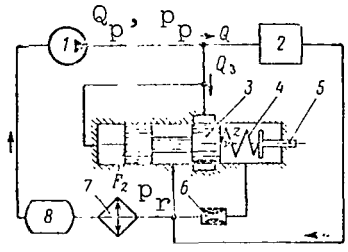

Figure 3.15. Schematic of hydraulic supply system with constant output pump and bypass valve:
1 — constant output pump; 2 — hydraulic load; 3 — bypass valve slide; 4 — bypass valve spring; 5 — adjusting screw; 6 — restrictor; 7 — oil-water radiator; 8 — tank

Depending on the flow rate required by the load, the supply pressure magnitude changes somewhat, which is a result of change of the spring compression and the hydrodynamic force on the slide as it moves. Taking as the null slide position that in which the slide slot overlap is equal to zero, we can obtain the following system of equations describing the subject hydraulic supply source (for $p_r = 0$): /75

$$\left.\begin{aligned} Q_p - k_{le} p_p &= Q + k_{V_1} \dot{p}_p + k_z z \sqrt{p_p} + A_z \dot{z}, \\ m\ddot{z} + f_z \dot{z} + C_z z &= A_z p_p - C_z z_0 - k_d k_z z p_p, \end{aligned}\right\} \qquad (3.19)$$

where Q_p is the pump output;

$k_z = \sqrt{\frac{2}{\varrho}} \mu b_z$; b_z is the width of the rectangular ports in the valve case;

$k_d = 0.62 \cdot 10^{-3}$ [11]; f_z is the damping coefficient, referred to the slide.

After linearizing these equations, we obtain the following system of hydraulic supply source equations, expressed in terms of increments relative to the regime (p_{p0}, Q_0, z_0):

$$\left.\begin{aligned} -k_z \Delta z = \Delta Q + k_{V_1} \Delta \dot{p}_p + (k_{p_1} + k_{1e}) \Delta p_p + \\ + A_z \Delta \dot{z}, \\ m \Delta \ddot{z} + f_z \Delta \dot{z} + (C_z + k_{dz}) \Delta z = (A_z - k_{dp}) \Delta p_p, \end{aligned}\right\} \qquad (3.20)$$

where $k_{p_1} = \frac{k_z z_0}{2\sqrt{p_{p0}}}$;

$$k_{dz} = \frac{\partial R_d}{\partial z} = k_d k_z p_{p0};$$
$$k_{dp} = \frac{\partial R_d}{\partial p_p} = k_d k_z z_0.$$

If we use a bypass valve with servoaction [24], i.e., a two-stage valve, we can usually neglect the inertia and damping of the ball operating in the first valve amplification stage. In this case the equations of the bypass valve with servoaction can also be reduced to a form analogous to (3.19) and (3.20).

The resulting system of equations describing the hydraulic supply source with constant output pump and bypass valve is analogous to the Equations (3.2) obtained for the hydraulic supply source with variable output pump. Therefore, the conclusions drawn in the preceding section also apply to the hydraulic supply system with constant output pump and bypass valve. /76

3.3. Influence of Working Fluid Properties on Hydraulic Drive Characteristics

Contamination of the working fluid has a major influence on hydraulic drive slide-valve pair friction forces, and also hydraulic drive plunger-pair friction forces. Contaminating particles whose dimensions are commensurate with the dimensions of the clearance

between slide-valve pair interacting elements, for example, between the slide and case, may enter this clearance together with the fluid stream and stick in this clearance. If there is a particle stuck in the clearance, as the moving element displaces there will be either destruction of the particle or scratching of the moving element surface by the stuck particle (depending on the relationship between the hardness of the particle and of the material of the slide-valve pair elements). This leads to considerable increase of the breakaway force. In the case of cylindrical slide-valve pairs with surface hardness HRC 58-62, and surface finish corresponding to lapping, for preliminary calculations we can find the maximal magnitude of the friction force F_f due to contamination of the fluid from the formula presented in [29]

$$F_f = (4 \cdot - 5)\pi d\delta\Delta p n_1$$

where d is the slide diameter; δ is the radial clearance between the slide and case; Δp is the pressure differential across the annular slot; n_l is the number of lands on the slide.

Increase of the friction forces in slide-valve pairs may also be caused by nonuniform fluid pressure distribution in the clearance between the slide and the case. This phenomenon, which leads to the appearance of unbalanced radial forces, has been analyzed in detail in [28]. We note that when using a slight taper on the slide lands in order to reduce pressure distribution nonuniformity, the friction force due to fluid contamination increases.

The slide breakaway force depends significantly on the time of /77
exposure to pressure and degree of working fluid contamination. This force increases with increase of the time the slide is stationary subjected to pressure. After holding for some definite time, the breakaway force reaches its maximal value and does not change with further increase of the holding time. Depending on the relationship between the pair dimensions, pressure differential across the slide valve slot, and degree of working fluid contamination, the duration $t_{F_{br}}$ of the breakaway force buildup process may vary from a few

minutes to an hour or more. For example, for the slide-valve distributor of the BU-10 booster with pressure 600 N/cm^2, and contaminant concentration 40 mg/liter, the breakaway force varies from 3 N to the maximal value 25 N after two minutes of exposure, while with carefully filtered fluid it varies from 3 N to 5 N after four minutes of exposure [29]. The typical range of the breakaway force variation with exposure time is shown in Figure 3.16.

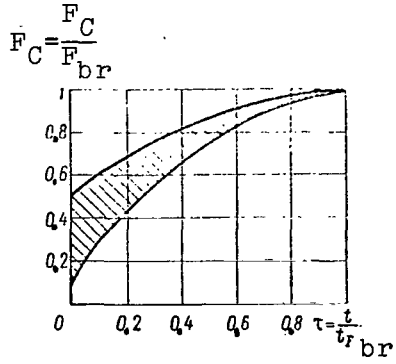

Figure 3.16. Slide breakaway force versus time under pressure

After slide breakaway, the slide displacement force usually decreases by several fold, which is explained basically by destruction of the particles as the slide moves and flushing of the particles from the clearance. Since the time during which the booster is under pressure in the stationary condition has considerable influence on the magnitude of the input member breakaway force, the exposure duration prior to measurement of this force must be indicated in the booster technical documentation. Depending on the booster operating conditions in the airplane control system, the required exposure time prior to measurement may vary from a few seconds to several hours.

The working fluid properties may have considerable influence on /78
the magnitude and nature of the leakage flow variation through the hydraulic drive as a function of time. The leakage flow in hydraulic drives is usually due to the resistance of the annular (or planar) slots of small section in the cylindrical (or planar) slides. The fluid flow rate through a concentric annular slot of width l for laminar flow is determined by the expression:

$$Q_{le} = \frac{\pi d \delta^3 \Delta p}{12 \mu l}. \qquad (3.21)$$

For eccentric positioning of the slide in the case, the leakage flow increases and for the maximal eccentricity, when the slide touches the case, it reaches 2.5 Q_{le}.

For a planar slot of height δ, the leakage magnitude can be determined from (3.21), where the slot perimeter πd is replaced by the slot width w (for $\delta << w$):

$$Q_{le} = \frac{w\delta^3 \Delta p}{12\mu l}. \qquad (3.22)$$

However, the leakage magnitude in slide valve pairs obtained in practice corresponds to the theoretical value [see (3.21) and (3.22)] only at the first instant after pressure application or after slide reversal, and then in the course of several minutes the leakage decreases markedly, and sometimes stops completely.

Reduction of the leakage flow rate through the capillary slide-valve slots is caused by two basic factors: blockage of the slot by mechanical particles present in the working fluid, and by obliteration. We note that blockage of the slot by mechanical particles is also sometimes considered obliteration however, these phenomena have a different physical nature and should be differentiated from one another.

Akhmatov showed in [1] that fluid flow in capillary channels is determined by the behavior and properties of the fluid in the boundary films which form on the surface of the solid walls. As a result of the action of electrical and intermolecular forces at the interface between the wall surface and the fluid,layers of polarized molecules are absorbed on the channel walls. These layers have definite mechanical strength and gradually reduce the slot cross section, and for small slots, may block them completely. /79

The intensity of slot blockage depends, for a specific working fluid condition, on the clearance dimension in the slide-valve pair, increasing with decrease of this clearance. For example, the results shown by the curves in Figure 3.17 [29] were obtained by running standard clean AMG-10 fluid through planar slots with clearances of 10, 15, and 21 microns. The most intense slot blockage occurs

with the 10 micron gap, while there is practically no gap blockage for clearances greater than 21 microns.

The leakage flow dependence on time should be considered in electing the technique for determining leakage in boosters and servo-controls, since results which differ by several fold can be obtained when using different techniques.

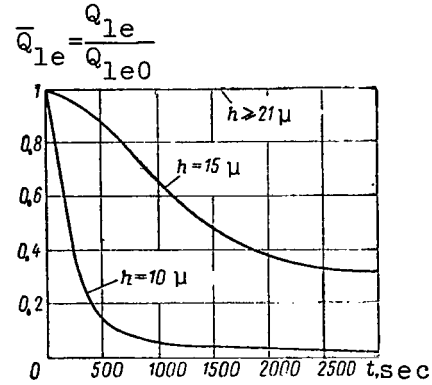

Figure 3.17. Leakage flow versus time for planar slots

Working fluid compressibility may have a marked influence on hydraulic drive dynamic properties, leading, in many cases, to unstable operation. The influence of compressibility may become still more critical if there is undissolved air in the working fluid. However, as was shown in Section 2.5, in the aircraft hydraulic drive operating pressure range, the presence of undissolved air in the working fluid does not have any marked influence on hydraulic drive dynamic properties.

Working fluid density influences slide-valve pair flow and hydraulic drive output member velocity, which decrease with increase of the fluid density. They may be very important when hydraulic drive operation using different fluids is necessary.

Working fluid viscosity has marked influence on the leakage /80
through hydraulic drive slide-valve pairs. During operation of the hydraulic system, "churning" of the fluid takes place, and this leads to reduction of the fluid viscosity because of the destruction of the viscous additives consisting of long hydrocarbon chains. In the hydraulic sppply system, with constant output pump and bypass valve, there is more intense "churning" of the oil than in the hydraulic supply system with constant output pump for the same maximal flow rate.

Sometimes it becomes necessary in practice to develop and produce hydraulic drives using different working fluids (for example, AMG-10 mineral oil and 7-50S-3 synthetic oil). In this case, it is advisable to have several test stands with different working fluids or periodically change the working fluid in the stands, since the operation of working fluid replacement in a stand is very time consuming. It is recommended that initial adjustment and acceptance-delivery tests of such hydraulic drives be made using one working fluid, and after terminating these tests, if necessary, another run and preservation of the hydraulic drive be accomplished using the other fluid. During the hydraulic drive tests, we must take into account the influence of the different physical and mechanical properties of the fluids (density, primarily) on the hydraulic drive parameters.

3.4. Test Stand Hydraulic Supply Systems and Hydraulic Instrumentation

As a result of examining the influence of the basic hydraulic supply system elements on hydraulic drive static and dynamic characteristics, we can draw the following conclusions.

1. During operation aboard an airplane, the hydraulic drive supply and return pressures are variable and depend on pump regulation characteristic slope, hydraulic line resistance, and the operating regimes of the other hydraulic system loads. However, since, in the technical specification, the requirements on parameter magnitudes are usually specified for constant supply and return pressures, /81
it is recommended that a system with constant output pump and bypass valve be used in the test stand hydraulic supply systems. This hydraulic supply system provides the simplest regulation of the required supply pressure magnitudes and precise maintenance of the established pressure magnitudes in variable regimes.

2. The nature of the influence of regulator inertia in the airplane hydraulic system variable output pump is analogous to the

nature of the influence of the test stand hydraulic system bypass regulating element on hydraulic drive dynamic characteristics. When using a stable hydraulic supply system and a hydraulic accumulator in the supply line, the degree of pressure regulating element inertia influence on hydraulic drive dynamic characteristics is negligibly small for both the airplane and test stand hydraulic supply systems. In the absence of a hydraulic accumulator in the supply line of the airplane hydraulic supply system with weakly damped or overdamped output regulator, the hydraulic supply system may have marked influence on hydraulic drive dynamic characteristics; however, such cases are encountered very rarely in practice. Thus the dynamic characteristics of hydraulic drives operating from the airplane hydraulic supply system (with variable output pump), or from the test stand hydraulic supply system (with constant output pump and bypass valve) are practically the same for a stable hydraulic supply system and a hydraulic accumulator in the supply line.

3. In the test stand hydraulic supply system, it is necessary to reproduce the same conditions in regard to working fluid contamination and its physical and mechanical properties as in the airplane system. Since in the test stand hydraulic supply system there is more intense "churning" of the working fluid than in the airplane system, the working fluid must be replaced more frequently in the stand.

4. The influence of pressure pulsations at the hydraulic drive inlet is negligibly small in both the airplane and test stand hydraulic systems.

5. The construction of the test stand hydraulic system and its elements should not permit excessive air content in the working fluid. Undissolved air content in the working fluid up to 1% does not have any significant influence on the hydraulic drive characteristics. /82

A schematic of a typical test stand hydraulic supply system is shown in Figure 3.18. One or more pumps 1 operating in in parallel are driven by the electric motors 2. To provide for the possibility of individual pump operation, check valves 3 are installed in their discharge lines. The working fluid enters the pumps from the tank 4, which is supplied with nitrogen under pressure (usually about 10 N/cm^2) to provide normal pump operation.

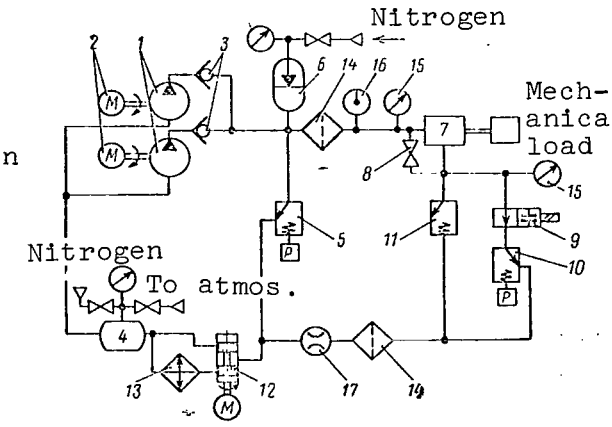

Figure 3.18. Schematic of typical test stand hydraulic system:

1 — pump; 2 — electric drive motor; 3 — check valve; 4 — tank; 5 — bypass valve; 6 — hydraulic accumulator; 7 — test hydraulic drive; 8 — bleed valve; 9 — selector valve; 10, 11 — relief valves; 12 — remotely controlled metering valve; 13 — oil-water radiator; 14 — filter; 15 — pressure gauge; 16 — temperature sensor; 17 — flowmeter

In order to regulate and maintain the required constant delivery pressure, the bypass valve 5 is installed in the supply line to bypass the excess fluid flow supply by the pump into the return line. The pneudraulic accumulator 6 may be installed /83 in the supply line. The adjustable reflief valve 10, which creates the required return pressure, is installed in the return line downstream of the test hydraulic drive 7. The bleed valve 8 is provided to ensure functioning of the valve 10 with nearly zero fluid flow rate through the hydraulic drive. The radiator 13, through which cold water flows constantly, is installed in the return line ahead of the tank to cool the working fluid and maintain the required temperature in the return line. Temperature regulation is accomplished by remotely controlled metering valve 12, which regulates smoothly the flow rate of the working fluid streams entering the tank directly or through the radiator.

A second relief valve 11, set to a maximal transient pressure exceeding the operating return pressure, is provided to dump excessive pressure in the return line. The valve 10 is installed downstream of the normally-open electromagnetically controlled selector valve 9. When an electric control signal is applied to close valve 9, valve 11 opens and provides the required pressure relief in the hydraulic drive return line. The duration of this pressure overshoot is determined by the time of electric signal application to the valve 9 and the dynamic properties of the selector 9 and valve 11. The valve 11 may also be used as a safety valve; in this case, its triggering pressure should exceed the operating return pressure by 10 - 15%.

Filters 14 are installed in the hydraulic supply system to ensure the required working fluid cleanliness. Usually, serge-weave wire gauze filters are installed in the supply and return lines to provide a 12 - 16-micron filtration level. When a 5-μ metalloceramic filter is installed in the airplane hydraulic system ahead of the hydraulic drive, a 5-μ filter is also installed in the hydraulic supply system pressure line. This ensures fluid cleanliness in the test stand hydraulic supply system close to the fluid cleanliness in the airplane hydraulic system.

In order to monitor hydraulic supply system operation and determine the required parameters of the hydraulic drives being tested, pressure gauges 15 are installed in the hydraulic supply system to measure the supply and return pressures, hydraulic accumulator charge and tank pressurization pressures; the temperature sensor 16 measures the working fluid temperature ahead of the hydraulic drive; and the flowmeter 17 measures the leakage flow rate or the hydraulic drive operating flow rate. /84

The hydraulic supply system scheme shown in the figure is usually used for operating fluid temperatures from +20 to +100° C. Special thermal test stands with working fluid heaters are used at higher temperatures.

The pressure in the hydraulic supply system is measured by indicating pressure gauges. Calibrated pressure gauges of accuracy class 0.5 - 0.6 (types MKO, MTI, and so on) are recommended for measuring the supply and return pressures, which define the hydraulic drive operating conditions. General purpose pressure gauges of accuracy class 1.6 — 25. (types OBM, M1V, MP4, and others) can be used to monitor hydraulic accumulator pressure and tank pressurization pressure. The pressure gauge measurement limits are selected so that the maximal measured pressure does not exceed two thirds of the maximal indications on the pressure gauge scale. Usually, the pressure gauge is connected to the hydraulic system through an adjustable restrictor to protect it against pressure pulsations and reduce pressure gauge pointer oscillations. The connection of the supply and return pressure gauge tubing to the supply and return lines should be at a point no further than one meter from the hydraulic drive being tested.

Fast-response strain-gauge pressure sensors are used to study the unsteady pressure variation processes and provide continuous recording of the pressure signal on a light-beam oscillograph. In the simplest strain-gauge pressure sensor, the sensitive element is a thin-wall cylinder which is deformed by the pressure and on which wire or foil strain gauges are bonded. The strain-gauge pickup internal volume (about 3 cm^3) usually has negligibly small influence on the dynamic processes in the hydraulic system.

The working fluid flow rate through the hydraulic drive is measured by flowmeters installed in the hydraulic supply system return line. The reason for this location is that most of the commercially produced flowmeters can operate only at low pressures. /85
Since the fluid volume increases with pressure reduction, the actual flow rate measured in the hydraulic drive return line will exceed the fluid flow rate entering the hydraulic drive by the magnitude $\Delta Q = \beta (p_{pu} - p_r) Q$.

With a pressure differential across the hydraulic drive of 2000 N/cm^2, and compressibility $\beta = 1/E = 7 \cdot 10^{-6}$ cm^2/N, the relative flow rate difference $\Delta Q/Q$ amounts to 1 - 3%. Usually, when measuring leakage flow rate (or utilized flow rate), it is required that the flow rate not exceed a definite magnitude; therefore, flow rate measurement in the return line, yielding a higher flow rate value in comparison with flow rate measurement in the supply line, is better. Fluid quantity counters with oval gears, for example, 2-SVShS-25 and others of accuracy class $\pm$ 0.5%, are widely used for measuring flow rate. Vane-type tachometric flowmeters (for example, RSS, "Saturn", and so on), rotameters (accuracy class 2.5), differential manometer flow meters (accuracy class 2.5), and others are also used.

Axial piston hydraulic motors connected with tachometric or impulse-type revolution counters are also widely used for measuring flow rates. When the hydraulic motors are installed in the return line, there are practically no leaks in the hydraulic motors, and the error caused by the motors is negligibly small. Most convenient for precise measurement of low flow rates is the use of cylindrical calibrated tanks which are connected by an electrically controlled slide valve with the hydraulic drive return line when it is necessary to make a measurement, and the tank piston begins to displace with a velocity proportional to the flow rate. The first microswitch triggers at the initiation of piston motion, and the second microswitch triggers as the piston approaches the upper piston. After the piston arrives at the extreme upper position, a valve in the piston opens, and the fluid passes through it into the return line to the tank. After closure of the control valve, the calibrated tank is disconnected from the return line, and the piston returns under the action of its weight to the original position. When making a measurement, the time (ΔT) for the piston to travel the distance between the two microswitches is recorded by an electric stopwatch. The working /86
fluid volume (ΔV) in the calibrated tank, corresponding to piston travel between actuation of the two microswitches, is determined from

preliminary calibration of the tank. The flow rate is found by dividing the constant tank volume ΔV by the measured time ΔT. The error of the flow rate measurment using the calibrated tank does not usually exceed $\pm$ 2%.

The working fluid temperature is usually measured by manometric or resistance thermometers operating together with balanced automatic bridges and logometers, and also by thermocouples operating together with millivoltmeters or self-recording potentiometers.

CHAPTER IV

TEST STAND LOAD SIMULATION SYSTEMS

4.1. Load Influence on Hydraulic Drive Characteristics

Constant and damping loads, and also the friction force on the /87
output member, may influence significantly hydraulic drive static characteristics. Constant opposing or aiding loads on the hydraulic drive output member lead, respectively, to decrease or increase of the output member velocity (see Figure 1.6). If a constant force acts with the output member stationary, it causes deflection of the booster output member (see Figure 1.8), and in the servocontrol it causes a null shift in the direction of load action, whose magnitude depends on the distribution slide-valve force characteristic slope.

A damping load leads to decrease of the hydraulic drive output member velocity. A friction force causes decrease of the output member velocity and also increase of the hydraulic drive deadzone, since in this case it is necessary to create a larger pressure differential and, consequently, larger displacement of the distribution slide-valve in order to start output member movement.

We note that an inertial load has no influence on hydraulic drive static characteristics, which are determined in steady-state operating regimes. However, in the presence of an inertial load,

comparatively lengthy transient processes, with respect to the output member velocity, may take place with application of a step-function control signal; therefore, when developing a technique for checking the output member velocity, it is necessary to make certain the measurement is made on the steady-state output member motion segment.

Let us examine further the influence of the primary loading /88 modes on hydraulic drive dynamic characteristics. The inertial load is one which affects most strongly the dynamic properties of hydraulic drives. It follows from analysis of booster [see (1.4)] and servocontrol [see (1.5)] transfer functions that the inertial load, together with working fluid compressibility, leads to the appearance of a resonance peak on the AFC. With increase of the inertial load, the resonance peak on the AFC increases and shifts toward lower frequencies, and the phase shifts on the hydraulic drive PFC increase. Increase of the inertial load leads to decrease of the amplitude and phase margins, and may cause loss of stability. The inertial load determines the oscillatory nature of the hydraulic drive transient processes, and with increase of the inertial load, the oscillation frequency decreases approximately as $M^{1/2}$, and overregulation increases. For servocontrols, with inertial load increase the velocity rise time constant T_{sc} increases approximately as $M^{1/2}$, and in this case the lag also increases somewhat because of increase of the "spoon-shaped" segment in the beginning of the transient process.

A positional load on the output member causes downward shift of the hydraulic drive AFC and some increase of the phase shifts, which increases hydraulic drive stability but reduces its response. Damping load and dry friction on the hydraulic drive output member also lead to increase of hydraulic drive stability and decrease of its response, and dry friction also reduces hydraulic drive static accuracy. However, we should note that ideal friction relay characteristics do not occur in practice, either in operation or under test stand conditions. In most cases, at low velocities there is a segment with negative damping, while at high speeds we observe positive

damping. The negative damping zone is most critical from the viewpoint of hydraulic drive stability, and in many cases leads to the onset of undamped auto-oscillations [35]. If the static friction of the output member load exceeds significantly the sliding friction, /89 output member jerking will take place at breakaway.

Hydraulic drive mounting elasticity may have a marked deteriorating influence on hydraulic drive stability, particularly in the presence of an inertial load. The influence of mounting elasticity is analogous to that of actuator working fluid compressibility increase, and shows up on reduction of the hydraulic drive resonant and natural frequencies. The influence of elasticity of the connection between the hydraulic drive output member and the load has a more complex nature [13]. This elasticity influences the resonant frequency in the same way as support elasticity. However, in addition to this, elasticity of the connection between the output member and the load introduces additional damping and a small static error.

4.2. Test Stand Load Simulation Systems*

The test stand loading systems (we shall use this term hereafter for the systems which simulate the load on the hydraulic drive output member) can be divided with respect to type of loading device into mechanical, hydraulic, pneumatic, and combined (pneudraulic). In the mechanical systems, the load is created by means of various springs, weights, and so on. In the hydraulic and pneumatic systems, the load is created by power cylinders, into the chambers of which the working fluid or compressed gas is supplied at the required pressure. With respect to operating principle, the loading systems can be divided into active and passive. The passive loading system does not have its own source of energy, and can create a constant or positional load on the hydraulic drive output member only with movement of the output member at the expense of energy of the hydraulic drive being tested. The active loading system has its own energy

*Written together with L. K. Dodonkin.

source (hydraulic pump, compressed gas bottle, compressed spring, and so on), and can create a load both with hydraulic drive output member movement and with the output member stationary. We shall examine the basic loading system schemes for simulating various types of loads (Table 4.1).

TABLE 4.1.

Load system type and loading mode	Constant		Positional	Inertial	Friction
	Unidirectional	Bidirectional			
Hydraulic					
a) active	⊕		⊕		
b) passive		⊕			
Pneumatic	+		+		
Combined pneumohydraulic	+		⊕		
Mechanical	+	+	+	⊕	⊕
Automated			+	+	

Constant load simulation systems

The constant load simulation systems can be divided into unidirectional and bidirectional, in which the direction of action of the constant load changes with reversal of the test hydraulic drive output member. Pneumatic and hydraulic loading systems have been most widely used to simulate a constant load. Two kinematic loading schemes have found use: with the loading cylinder located on the same line with the hydraulic drive (Figure 4.1a), and with parallel positioning of the loading cylinder (Figure 4.1b). The second scheme permits smaller test stand dimensions, but the magnitude of the "parasitic" added mass for this scheme is larger because of the presence of the massive pivoting lever. /90

Wide use has been made of loading systems with a hydraulic cylinder in one chamber of which the required differential pressure is created, and the pressure in the other chamber is the return

pressure. The schematic of an active loading system with unidirectional constant loading is shown in Figure 4.2. In this system a pump driven by an electric motor supplies to the corresponding hydraulic drive chamber, fluid at the required pressure, which is established by a bypass valve. A radiator cooler is installed ahead of the tank to prevent overheating of the fluid in the system during extended operation. The selector valve 8, which connects the pump supply (and return) line with the corresponding loading cylinder chambers is actuated to change the direction of load action. By actuating the selector 8 when the hydraulic drive output member reaches the extreme position, we can use this scheme to create a reversible constant load. The load force for this system is

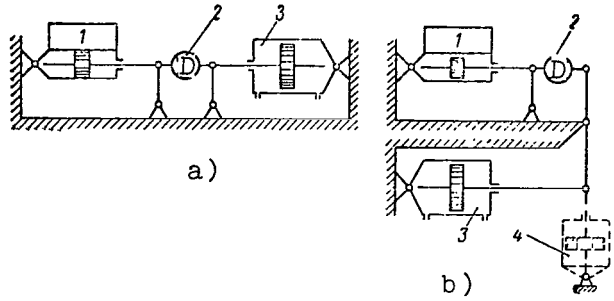

Figure 4.1. Kinematic schemes of loading stands:

a — with in-line positioning of loading cylinder; b — with parallel positioning of loading cylinder; 1 — test hydraulic drive; 2 — dynamometer; 3, 4 — loading cylinders for creating constant and positional loads

/91

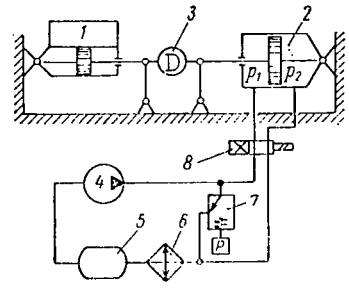

Figure 4.2. Schematic of active hydraulic system for simulating unidirectional constant load:

1 — test hydraulic drive; 2 — cylinder hydraulic loading; 3 — dynamometer; 4 — pump; 5 — tank; 6 — oil-water radiator; 7 — bypass valve; 8 — selector valve

$$R = A_{cyl} (p_1 - p_2) \pm R_{fr}$$

where A_{cyl} is the active loading cylinder piston area; p_1, p_2 are the pressures in the cylinder supply and return chambers, R_{fr} is the friction in the hydraulic drive. The sign "+" corresponds to an opposing load on the hydraulic drive, the sign "-" corresponds to an aiding load.

The pressures p_1 and p_2 differ from the supply pressure at the pump outlet and from the pressurization pressure, respectively, because of the losses in the supply and return lines. The magnitude of the opposing load decreases with decrease of the hydraulic drive output member velocity, and reaches a minimum for zero velocity. The magnitude of the cylinder force change, with change of the velocity from zero to the maximal value (v_{max}), is /92

$$\Delta R = A_{cyl}^3 (r_s^2 + r_r^2) v_{max}^2$$

where r_s and r_r are the hydraulic resistances of the loading system supply and return lines, which are determined by the losses along the length of the lines, local losses in the line fittings, in the selector valve, and also in fluid entrance into and exit from the cylinder. For high-speed hydraulic drive loading systems, the ratio $\Delta R/R_{max}$ reaches 10 - 15%. As a result of good seal lubrication conditions in the hydraulic cylinder and the possibility of operating at high pressures (up to 2000 N/cm^2), the hydraulic cylinder internal friction usually does not exceed 3 - 5% of the magnitude of the load created, and the no-load friction (in the absence of differential pressure in the cylinder chambers) does not exceed 1 - 2%. The primary drawback of the active hydraulic loading system is its relative complexity, size, and the necessity for using electric motors which require considerable electric power.

In the simplest passive hydraulic loading system scheme, the reversible constant load is created by interconnecting the cylinder chambers through a controllable restrictor valve. In this case, the load is created because of the appearance of a pressure drop (Δp) across the restrictor valve with passage through the valve of the flow rate $Q = A_p v$. In this loading system, the (opposing) load force depends on the hydraulic drive output member velocity and the set controllable magnitude of the restrictor valve resistance r_r

$$R_{op} = A_{cyl}^3 v^2 r_r^2 + R_{fr}.$$

This system can be used only for short-term operation with low flow rates and forces, i.e., for low power dissipated by the loading system. If there is significant dissipated power, the fluid in the /93
loading system heats up rapidly, which limits the time of continuous operation and leads to instability of the established load magnitude. Moreover, the presence of load "slop", which arises because of air getting into the hydraulic loading system when it is filled or because of imperfect sealing of the system, is characteristic of this loading system.

Figure 4.3 shows two improved passive loading schemes (A and B) which permit long-term operation with accurate maintenance of the established reversible load and automatic maintenance of the established load with the use of relief valves. In both schemes, check valves are used to permit the installation of a cooling line, in which high pressures do not develop regardless of the direction of load force action. A bypass valve is provided in both schemes to permit the creation of low loads and free displacement of the loading cylinder piston. The bypass valve is required, since the relief /94
valve provides a minimal pressure of 50 - 150 N/cm^2 in the loading system. Scheme A does not have a pressurized tank and, therefore, air can get into the system if sealing is not perfect and increase the load "slop". As a result of use of a pressurized tank in scheme B, the influence of imperfect hydraulic system sealing has considerably less effect. Scheme A provides equality of the forces with loading cylinder piston motion in both directions, while in scheme B, forces which differ in magnitude may be created with piston motion in opposite directions by different adjustment of the relief valves.

Let us examine load variation as a function of time and piston travel for the active and passive hydraulic loading systems with constant relief valve setting (set to p_{1r}). Figure 4.4 shows a cyclogram of output member motion and hydraulic drive load variation /95
for step-function control signal change. For the active loading system, prior to initiation of hydraulic drive piston rod movement from the extreme position, the load is $R_0 = A_p p_{1r} - R_{fr}$ (with $p_2 = 0$).

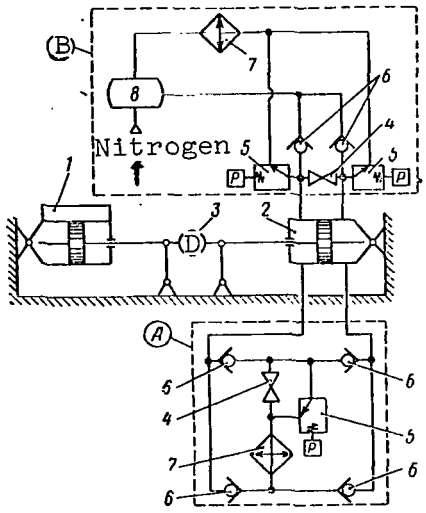

Figure 4.3. Schematic of passive hydraulic system for simulating reversible constant load:

1 — test hydraulic drive; 2 — hydraulic loading cylinder; 3 — dynamometer; 4 — bypass valve; 5 — relief valve; 6 — check valve; 7 — oil-water radiator; 8 — tank

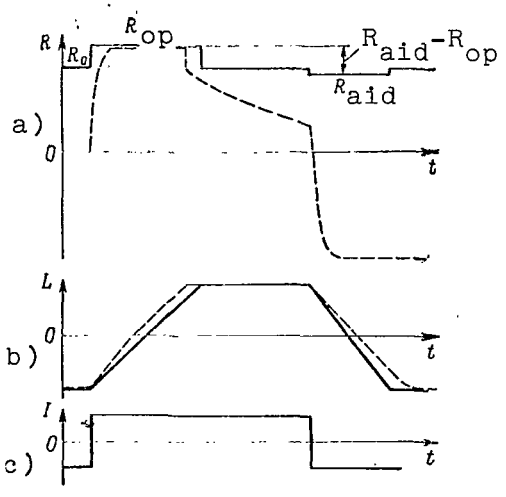

Figure 4.4. Cyclograms of test hydraulic drive control signal variation (c), output member motion (b), and output member load (a):

——— — active hydraulic loading system; - - - - —passive hydraulic loading system

After motion initiation, there arises partically instantaneously the opposing force R_{op}, which exceeds the value of R_0 by twice the friction in the hydraulic cylinder piston, plus the force resulting from pressure losses in the supply and return lines:

$$R_{op} = A_{cyl} (p_{lr} + \Delta p_d) + R_{fr}.$$

After the hydraulic drive piston rod reaches the extreme position, the load force R_0 is again established. In this case, the specified constant load R_{op} acts nearly throughout the entire hydraulic drive piston rod travel.

With reverse movement of the hydraulic drive piston, the load becomes aiding, and its magnitude is:

$$R_{aid} = A_{cyl} (p_{lr} - \Delta p_d) - R_{fr}.$$

Thus, for the same loading system relief valve setting, the difference between the opposing and aiding load magnitudes is equal to the sum of twice the friction forces in the hydraulic loading cylinder and the force resulting from hydraulic losses in the supply and return lines:

$$R_{op} - R_{aid} = 2A_{cyl}\,\Delta p_d + 2R_{fr}.$$

In the passive loading system, the load force is zero prior to initiation of hydraulic drive operation. After initiation of hydraulic drive output member movement, the magnitude of the opposing load increases smoothly to the required magnitude R_{op}.

The load rise time depends on the amount of undissolved air in the hydraulic loading system. Since it is not possible in practice to remove all the undissolved air from the hydraulic loading system, the load rise zone usually amounts from 0.1 to 0.3 of the hydraulic drive piston travel. In this segment, the hydraulic drive piston rod travels with higher speed than with a constant load. After the hydraulic drive piston rod reaches the extreme position and stops, the /96
fluid pressure in the high-pressure chamber begins to decrease because of leaks in the relief valves, and the load on the hydraulic drive piston rod also decreases. The load reduction time depends on the degree of hermeticity of the closed valve. In the case of complete valve hermeticity, the load existing at the end of load piston travel is maintained for a long time, and the bypass valve must be cracked open to bleed it off.

We note that, when reversing the hydraulic drive output member in the passive loading system at the beginning of the travel, in the course of a few tenths of a second, there may be an aiding load due to the influence of the residual pressure in the loading cylinder chamber.

In many cases, an active pneumatic loading system, which is considerably simpler than the hydraulic loading system, is used to

create relatively small loads (no more than 15 kN). A schematic of a pneumatic loading system is shown in Figure 4.5. The required pressure is created in one chamber of the pneumatic loading cylinder by the high-pressure bottle 8 and a pressure reducer. After establishing the required pressure, the shutoff valve 5 is closed. The high-pressure chamber in the pneumatic cylinder is connected by a line of large cross section with the bottle 4, which serves to maintain a constant load force as the hydraulic drive piston moves. The valve 6 is used to vent the gas pressure from the high-pressure chamber. The second chamber of the cylinder is connected to ambient air. The opposing force developed by the pneumatic loading cylinder is equal to $R_{op} = A_p p_1 + R_{fr}$, and the aiding force is $R_{aid} = A_p p_1 - R_{fr}$. For pneumatic cylinders, the relative magnitude of the friction may reach 10 - 15%, exceeding by about three times the relative magnitude of the friction in the hydraulic cylinder. With sufficiently large and short lines connecting the pneumatic cylinder working chamber with the bottle, we can consider that the gas pressure in the cylinder is independent of piston velocity, and varies as a function of piston position in the pneumatic cylinder (considered equal to the test hydraulic drive output member travel) as follows:

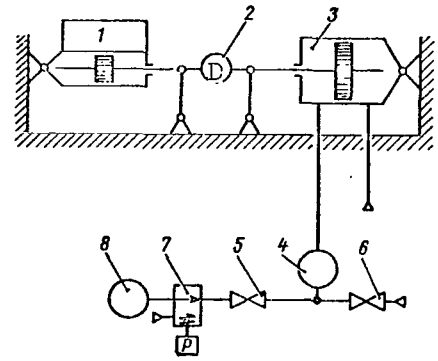

Figure 4.5. Schematic of pneumatic loading system:
1 — test hydraulic drive; 2 — dynamometer; 3 — pneumatic loading cylinder; 4 — bottle; 5, 6 — shutoff valves; 7 — pressure reducer; 8 — high pressure bottle

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$$p_1 = \frac{p_0}{1 \pm \frac{\Delta V}{V_0}} = \frac{p_0}{1 \pm \frac{A_{cyl}\, y_{max}}{V_0}\,\bar{y}},$$

where p_0 is the gas pressure for neutral pneumatic cylinder piston position; V_0 is the gas volume for the neutral piston position; $\Delta V = A_{cyl}\, y_{max}\, \bar{y}$; $\bar{y} = \frac{y}{y_{max}}$ is the relative piston displacement; $\Delta V_0 = A_{cyl}\, y_{max}$ is the chamber volume change with maximal piston displacement.

From this equation, we can find that, in order to maintain the load within $\pm$ 5% as the piston travels, we must use a bottle volume for which $V_0 > 20\ \Delta\ V_0$.

Drawbacks of the pneumatic loading system are high friction and short service life of the pneumatic cylinder due to the poor pneumatic cylinder piston seal lubrication conditions.

Mechanical loading systems can also be used to create constant loads. For small loads (up to 1 kN), we can use weight attached to the piston rod by means of a cable. However, in this case, the inertial load has a marked influence. This leads to large shock accelerations at the end of the travel as the hydraulic drive output member comes to a stop. /98

A constant opposing reversible load can be created with the aid of a passive loading system with friction, which is examined in more detail somewhat later. The drawbacks of this technique for simulating a constant load are the impossibility of creating large loads (more than 2 kN), considerable heating of the friction pair during operation, rapid wear of the friction pair, instability of the established friction magnitude during long-term operation, and dependence of the force on the velocity.

Positional load simulation systems

Various loading devices of the mechanical type utilizing springs are widely used to simulate positional loads on the hydraulic drive output member. Figure 4.6 shows a loading device design with

cylindrical spring. In order to obtain different positional load stiffness, various interchangeable springs, each of which has its own interchangeable follower, are installed in the loading cylinder.

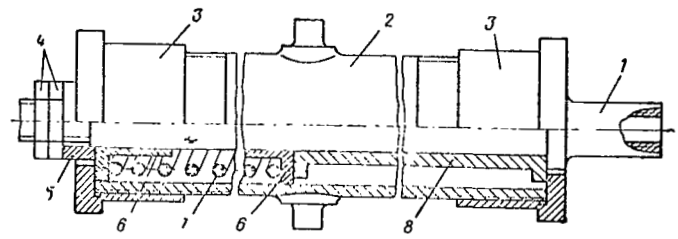

Figure 4.6. Spring device for simulating positional load:

1 — piston rod; 2 — cylinder; 3 — nut; 4 — lock nut; 5 — ring; 6 — bushing; 7 — spring; 8 — follower

Various hydraulic and pneumatic systems are also used to simulate a positional load. One of the widely used simulation techniques is the introduction (see Figure 4.1b) of a vertical hydraulic or pneumatic cylinder which creates a constant /99 force. In the neutral position of the hydraulic drive output member, the cylinder is coaxial with the pivoting lever, and the force created by the cylinder is absorbed by the lever shaft, and is not transmitted to the hydraulic drive output member. As the hydraulic drive output member displaces from the neutral position, the lever rotates and the cylinder is positioned at an angle to the lever, as a result of which the tangential component of the cylinder force creates on the output member a force which varies approximately in proportion to its displacement from the neutral position. The magnitude of the pressure in the cylinder determines the slope of the positional load. We note that the positional load created in this fashion has hysteresis due to the influence of friction in the cylinder. This scheme is particularly convenient for use when simultaneous simulation of constant and positional loads is required. In this case, two mutually perpendicular force cylinders act simultaneously on a single pivoted lever.

Sometimes use is made of a hydraulic positional load simulation system analogous to the scheme of Figure 4.2, but with automatic regulation of the valve which controls the pressure in the hydraulic cylinder chamber. Control of the valve can be accomplished by a

cam mechanism driven by an auxiliary device from the control signal or from the hydraulic drive output member.

Pneumatic systems have not found wide application for positional load creation. In the simplest differential pneumatic system (pneumatic spring), both cylinder chambers are charged in the neutral piston position with gas under a definite pressure. However, this system has the drawbacks noted above, which are characteristic for pneumatic systems.

Sometimes use is made of the loading scheme with pneumatic spring or pneumatic chamber (for example, from the airplane landing gear). In this case, the hydraulic drive output member is connected with a pivoting lever on the end of which the pneumatic chamber is mounted. In the lever position corresponding to the hydraulic drive /100
output member neutral position, the pneumatic chamber is constrained on both sides by stationary stops. As the hydraulic drive output member displaces from the neutral position, the gas in the pneumatic chamber is compressed by the corresponding stop, creating an opposing force approximately proportional to the displacement from the neutral position. The slope of the positional load can be varied by regulating the gas pressure in the pneumatic chamber, and by varying the lever arms. The drawback of this scheme is the presence of considerable "parasitic" inertial load on the hydraulic drive output member.

The active differential combined pneudraulic positional load simulation system shown in Figure 4.7 is very convenient. The load on the hydraulic loading cylinder piston is created as a result of the working fluid pressure differential caused by unbalance of the compressed gas pressures in the hydraulic accumulators with hydraulic drive output member displacement in either direction from the neutral position, at which the gas pressures in the two accumulators are equal. A definite temperature is maintained in the hydropneumatic system with the aid of the radiators 4. We note that a low-power /101
cooler is required in this system, and in many cases the hydraulic system can operate normally without a cooler. The gas chambers of

the hydraulic accumulators are charged with the aid of the compressed gas bottle, gas reducer, and valves 12, to a pressure which is monitored by the pressure gauge 13.

The hydraulic accumulator hydraulic chambers, loading cylinder chambers, and the connecting lines are filled with the working fluid at the required pressure with the aid of an auxiliary hydraulic system, consisting of a pump, working fluid tank, valves 8, and relief valve 5.

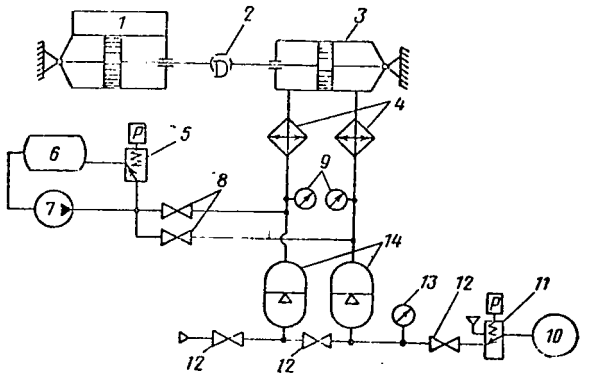

Figure 4.7. Schematic of combined pneumatic-hydraulic positional load simulation system:

1 — test hydraulic drive; 2 — dynamometer; 3 — hydraulic loading cylinder; 4 — radiator; 5 — relief valve; 6 — tank; 7 — pump; 8, 12 — shutoff valves; 9, 13 — pressure gauges; 10 — high-pressure bottle; 11 — reducer; 14 — hydraulic accumulator

In view of the practical importance of this case, we shall calculate the parameters of the subject loading device with piston-type pneudraulic accumulators. Considering the processes in the gas chambers to be isothermal, we can take for a definite gas mass

$$p_0 V_0 = p_c V_c, \qquad (4.1)$$

where p_c and p_0 are the charge pressure and working pressure for neutral piston position ($y = 0$), V_c and V_0 are the volumes of each gas chamber during charging and operation (for $y = 0$). To simplify the calculations, we assume that there is no friction in the loading cylinder or in the hydraulic accumulators.

When calculating the loading device parameters, the initial values are usually the maximal hydraulic drive output member load R_{max} and the maximal output member displacement amplitude y_{max}.

We assume in the calculations that $p_1 \geq p_2$, where p_1 and p_2 are the instantaneous pressures in each of the hydraulic cylinder chambers. It follows from (4.1) that:

$$p_1 = \frac{p_c V_c}{V_0 - \Delta V}; \quad p_2 = \frac{p_c V_c}{V_0 + \Delta V}.$$

Then the pressure differential in the hydraulic cylinder is:

$$\Delta p_{cyl} = p_1 - p_2 = \frac{2p_c V_c \Delta V}{V_0^2 - \Delta V^2}.$$

Since $\Delta p_{cyl} = \frac{R}{A_{cyl}}$, the equation for the hydraulic drive output member load can be obtained in the form: /102

$$R = \frac{2p_c V_c A_{cyl}^2}{V_0^2 - A_{cyl}^2 y^2} = \frac{2p_c V_c y}{\sigma^2 y_{max}^2 - y^2}, \qquad (4.2)$$

where $\sigma = \frac{V_0}{\Delta V_{max}}$.

The characteristics of the loading device elements can be selected on the basis of the limiting values R_{max} and y_{max}. Usually, in selecting the characteristics of the loading device elements, it is required that the maximal pressure $p_{1\ max}$ not exceed the limiting value p_{lim}, which is due to the structural strength of the pneudraulic system elements. Then,

$$\frac{p_c V_c}{V_0 - \Delta V_{max}} = p_{op}; \quad \frac{p_c V_c}{V_0 + \Delta V_{max}} = p_{op} - \Delta p_{max}.$$

Hence, we can obtain:

$$\Delta p_{max} = \frac{2p_{op}}{\sigma + 1};$$

or

$$A_{cyl} = \frac{R_{max}(\sigma + 1)}{2p_{op}}.$$

It follows from these expressions that it is possible to obtain the largest pressure differential across the loading cylinder piston and the smallest active loading cylinder piston area (which plays a major role in loading device design) with the smallest values of the coefficient σ. The smallest value of the coefficient σ which is acceptable in practice is σ = 1.5. After selecting the magnitude of the hydraulic cylinder active area, we can find the quantity V_0:

$$V_0 = \sigma \Delta V_{max} = \sigma A_{cyl} y_{max}$$

The quantities V and p_{ch} can be found from the expression:

$$p_{ch} V_{ch} = p_{op} \Delta V_{max} (\sigma - 1).$$

The choice of the quantities p_{ch} and V_{ch} is determined by the technical facilities which the loading device designer has available. /103 If the charging equipment (compressor, centralized air line, or compressed gas bottle) is the determining factor, after selecting the pressure p_c, we can determine the required volume of the accumulator or battery of accumulators. If when designing the loading device, the primary requirements are those of the configuration and, consequently, the dimensions of the hydraulic accumulators, we specify the hydraulic accumulator volume and determine the quantity p_c.

After selecting the quantities A_{cyl}, V_0, p_c, and V_c, we can plot a graph of the relation R = R (y). In the general form, this relation is nonlinear; however, for $\Delta V_{max} \ll V_0$, this relation is nearly linear, while for $\Delta V_{max} \approx V_0$, it is quadratic.

Thus, the magnitude of the coefficient σ determines the linearity of the relation $R = \phi(y)$, and to increase the linearity, we must increase σ, which contradicts the previously obtained requirement for reduction of σ. It is obviously necessary to determine the optimal value of σ for which satisfactory linearity of the load characteristic is achieved without excessive increase of the loading device dimensions. The maximal deviation of the value of R from linearity will occur for $y \approx 0.65\ y_{max}$. Then the nonlinearity of the loading load characteristic can be described by the coefficient γ:

$$\gamma\% = \frac{0{,}65R_{max} - R_{0{,}65}}{R_{max}} \cdot 100,$$

where $R_{0.65}$ is the load for $y = 0.65\ y_{max}$.

Substituting (4.2) into this expression, we obtain

$$\sigma = \sqrt{\frac{37{,}5 + 0{,}42\gamma}{\gamma}}.$$

Hence, we can find that $\sigma > 2$ is required to achieve nonlinearity $\gamma \leq 10\%$, and $\sigma > 2.8$ is required to achieve $\gamma \leq 5\%$. Since usually the allowable load nonlinearity does not exceed 10%, we can recommend using in practice the range of values $\sigma = 2 - 3$.

We note that in this scheme, the auxiliary pump setup can be low-power, and is connected to the test stand only to charge the hydraulic system. An additional advantage of this system is the possibility of creating, with the aid of the pump setup, any required pressure in the pneumatic system without the limitations imposed by the available compressed gas sources. /104

Inertial load simulation devices

To simulate an inertial load (equivalent mass) on the hydraulic drive output member, use is made of schemes with translational movement of a mass and with rotational motion of a lever with weights,

whose moment of inertia corresponds to the required reduced mass magnitude. Schemes with translational motion of a mass have not been widely used because of inconvenience in operation and the inevitable presence of "parasitic" friction or a constant force. The most widely used inertial load simulation scheme is shown in Figure 4.8. The construction of the levers and weights makes it possible to install the weights on any lever arm, which makes it possible to obtain easily the required magnitude of the referred mass M_{ref} by determining the weight installation lever arm, with aid of the expression:

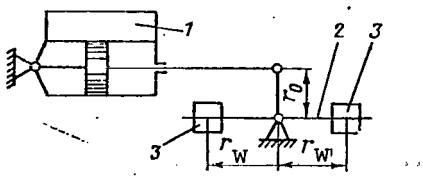

Figure 4.8. Schematic for simulating inertial load:
1 — test hydraulic drive;
2 — rocker with levers;
3 — weights

$$M_{ref} = \frac{2P_w r_w^2}{r_0^2},$$

where P_w is the mass of each weight, r_0 is the lever arm of hydraulic drive output member action on the pivoting rocker and levers; r_w is the weight installation lever arm.

This scheme is very convenient for the test stand configurations shown in Figure 4.1b, since the shaft of the rocker with inertial load simulator can be connected with the shaft of one of the test stand bellcranks.

Device for simulating friction force

In many cases, particularly when recording hydraulic drive dynamic characteristics, we are required to simulate output member friction. Friction pairs with translational and rotational motion are used to simulate friction. In the translational friction pairs, the actuator piston rod is connected with a moving plate, while the other stationary plate is forced against the moving plate with the

/105

aid of springs, in the direction perpendicular to the direction of motion. The clamping force and, therefore, the friction force are regulated by spring tension. In the rotating friction pairs, brake shoes are forced with the aid of spring devices against a rotating disk, which is connected with the hydraulic drive output member. The rotational friction pairs usually permit obtaining simpler and more compact devices for simulating the friction force.

When recording the dynamic characteristics, we must first verify carefully the actual friction force dependence on hydraulic drive output member velocity, and not permit an excessive drooping segment of this relation. In dynamic checks of hydraulic drives, it is very important to simulate the given elasticity of the support and the connection between the output member and load. For example, to simulate support elasticity, we can mount the unit either on a rigid intermediate plate connected with the test stand support frame with the aid of a bidirectional adjustable set of spring washers, or on the central part of a square or rectangular flexible plate, attached long the ends to the test stand support frame, or on a cantilever vertical flexible bracket which makes it possible to mount the unit on different lever arms. A link with a spring of given stiffness is installed to simulate the elasticity of the connection between the test hydraulic drive and the inertial load.

Automatic loading systems, a schematic of one of which is shown in Figure 4.9, can be used to create loads of various types. The output member load is created with the aid of a mechanical spring, analogous to that shown in Figure 4.6. The required load magnitude is established with the aid of an electrohydraulic servomechanism /106 which controls the position of the spring unit housing. With the aid of test hydraulic drive output member position, velocity, and acceleration signals, the summer forms the signal u_R, characterizing the instantaneous required output member force magnitude. With the spring loading device, there is coupled the output member force sensor, which forms the electrical force signal u_{R_ϕ}. The error signal

$\Delta u_R = u_R - u_{R_\phi}$ controls the servomechanism piston rod position, reducing this load error to zero. This load simulation system is very complex and expensive and is advisable for use only when it is necessary to alter the nature and magnitude of the load frequently.

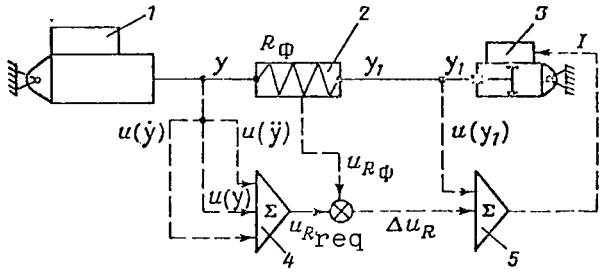

Figure 4.9. Schematic of automatic loading system:

1 — test hydraulic drive; 2 — spring loading device; 3 — electrohydraulic control mechanism; 4 — summing amplifier which forms the signal corresponding to the required load force; 5 — servomechanism drive summing amplifier

Dynamometers with elastic elements whose deformation is measured either by mechanical devices or by strain-gauge sensors have been widely used to measure the output member forces. Conventional dial indicators can be used as the mechanical device for measuring the elastic element (yoke or ring) deformation. For strain-gauge force measurement, wire or foil strain gauges are bonded to the elastic element and connected in a bridge circuit. Mechanical dynamometers are simple and reliable, but they have considerable inertia, and their use is advisable only for determining static characteristics and for output member oscillation frequencies up to 0.5 Hz. Dynamometers with strain gauges are practically inertialess and make possible automatic recording and remote measurement of the force.

Thus, for simulation of constant and positional hydraulic drive output member loads, we can recommend the hydraulic or pneudraulic loading systems, which provide high accuracy of required load reproduction, and also high stiffness and long operating life in comparison with the pneumatic system. The somewhat greater complexity of the hydraulic load simulation systems does not present any difficulties for hydraulic drive developers, since we can use for these

systems the same standard components as for the test hydraulic drive hydraulic supply systems. The recommended loading system types for each required loading mode are indicated by stars in Table 4.1.

CHAPTER V

HYDRAULIC DRIVE TESTING UNDER STANDARD CONDITIONS

5.1. General Test Requirements

One of the most important stages of hydraulic drive stand testing are the tests under standard conditions, for which the external factors are defined by the Expression (2.3). All the required hydraulic drive parameters are usually verified during these tests. We note that in many cases, those parameters for which the influence of high or low temperatures is more critical (maximal leakage, for example) are often not checked in the tests under standard conditions. /108

The techniques for verifying the purely electrical parameters of electrohydraulic units (check of electrical circuit resistance, insulation resistance and strength, and so on) do not differ from the techniques for the analogous checks of electronic instruments [8, 15, 25], and are not examined in the present study.

Weight is very important for all aircraft hydraulic drives. Weight is checked with the hydraulic drive completely filled with working fluid. During weighing, the hydraulic unit fittings are capped off.

A general requirement when conducting the tests is operation with a fluid which meets the requirements outlined in Section 2.5. It is recommended that working fluid cleanliness be analyzed daily, and analysis of the working fluid physical and chemical properties be made every 100 hours of test stand operation.

Hydraulic drives are tested on stands which include a hydraulic supply system and a system for loading the output member of the hydraulic drive being tested. Typical schematics of these systems were /109
examined in Sections 3.4 and 4.2. The construction of the stand may vary. Since the pump units create considerable noise during operation, they should be located in a separate sound-insulated room. The pump unit usually includes electric drive motors, pumps, cooler, and tank. All the hydraulic and pneumatic monitoring, metering, and regulating apparatus should be arranged on the hydraulic control console. The hydraulic drive being tested and the equipment used to load its output member are located on a structural frame. The hydraulic control console and structural frame should be made in the form of separate units which are mounted side by side and interconnected by hydraulic lines. The hydraulic control console is connected with the pump unit by hydraulic lines. This modular test stand configuration makes it possible to create comparatively simple construction of each of the stand modules and to standardize these modules. The test stand equipment complex also includes devices and equipment necessary to determine the required hydraulic drive characteristics and create the specified operating conditions.

5.2. Booster Test Technique

Verification of input and output member travel and maximal error

In order to determine booster input and output member travels, a force F_0, exceeding by 1.5 to 2 times the maximal breakaway force, is applied alternately in both directions to the input member by means of a weight, cable, and pulleys, with the hydraulic supply

connected and operating. The differences of the metering instrument indications corresponding to the extreme positions yield the booster input and output member travel magnitudes.

The order to determine the maximal error, a force is applied as indicated above to the input member alternately in opposite directions, with the booster output member stationary in a position close to neutral, and with the hydraulic supply turned off. In this case, the input member displacements will be limited by the slide valve distributor stops. Half of the input member position difference obtained defines the maximal booster error. We note that for booster transmission coefficient $k_t = y/y_{in} = 1$, the input member travel exceeds the output member travel by twice the maximal error. When conducting these checks and, also, all the subsequent booster checks, the input member travel is measured along the chord of the arc described by the airplane control system input link axis. The force on the input member is also applied to this axis. /110

Check of breakaway force and dead band

To conduct these checks, the output member is fixed in an intermediate position (any position off the stops). Since the breakaway force magnitude depends strongly on the booster dwell time (see Section 3.3) in the stationary condition with the hydraulic supply turned on, during the tests we should pay attention to the actual duration of the dwell time prior to initiating the breakaway force check. If the breakaway force, after dwell duration T (for example, 5 or 60 seconds), is specified in the technical documentation, then the measurement should be made within no more than 20 seconds after dwell for the required time. If the force measurement is not made during this time period, the output member should be deflected through full travel two or three times, after which the check is repeated. In those cases in which the required dwell time exceeds 30 minutes, the measurement duration need not be limited. In order to measure the breakaway force, after the required dwell time the force necessary for displacement of the output member at a velocity of no

less than 5 mm/min should be applied smoothly to the input member with the aid of a spring dynamometer or weight. The output member velocity can be determined with the aid of a dial indicator. The breakaway force is usually checked alternately for output member movement in both directions.

The dead zone can be checked by several techniques, depending on the booster peculiarities and the requirements imposed on the booster. The simplest and fastest method for checking the dead zone /111
is based on booster functioning for a given input member displacement. After setting the output member in an intermediate position, the hydraulic supply is turned off, and with the aid of a fixture, the input member displacement is limited to the maximal allowable dead zone magnitude. Then the hydraulic supply is turned on and the input member is displaced in the limits of the established constraints. If as the input member displaces, the output member also displaces, the booster dead zone does not exceed the allowable value. The output member displacement is determined with the aid of a dial indicator. The drawback of this technique is the impossibility of obtaining the exact dead zone magnitude. We note that a modification of this technique is possible when it is necessary to measure the actual dead zone magnitude. To accomplish this, we perform the check outlined above several times, gradually reducing the established input member limitations until output member displacement terminates.

Another method involves checking the dead zone on the basis of output member velocity. In this case, with the hydraulic supply turned on, the input member is alternately displaced relative to the selected stationary intermediate position in first one and then the other direction by the magnitude of the travel required to obtain an output member velocity of 5 ± 3 mm/min. The dead zone is determined from the difference of the two input member positions obtained, measured using a dial gauge. The drawback of this technique is the necessity for using a comparatively complicated fixture, which must be coupled with the input and output members, measure the input member travel relative to the output member travel (error), provide

smooth regulation of the error, and must also be adequately rigid. In certain cases, particularly for boosters with complex kinematics, the construction of such a fixture is a quite complex problem. Experience shows that the recommended output member velocity 5 $\pm$ 3 mm/min, which determines the dead zone boundary, is the minimal stable output member velocity. /112

The dead zone checking technique based on output member position (using the hysteresis loop due to influence of the dead zone) makes it possible to obtain the highest accuracy of actual dead zone magnitude determination. In accordance with this method, smooth displacement of the input member in one direction by a magnitude of about three times the maximal allowable dead zone ($y'_{in} \approx 3y_{dz}$) is accomplished with the aid of a special attachment relative to any intermediate position which is taken as the null position; then the input member is displaced in the opposite direction, relative to the original position, by the same magnitude and back again to the original position. The positions of the input and output members are recorded every (0.5 $\pm$ 0.2) y_{dz} with displacement of the input member. Then these measurements are used to plot the hysteresis loop of output member displacement y as a function of input member displacement y_{in}. The width of the hysteresis loop characterizes the dead zone, which can be determined from the arithmetic mean of three values of the loop width for output member displacements 0 and $\pm$ 0.5 y', where y' is the output member displacement for y'_{in}. The input and output member displacements are determined using dial gauges. In order to reduce the influence of obliteration and silting of the slide-valve ports by contaminant particles, the duration of the dwell time during measurement at each steady state position should not exceed 20 seconds.

The drawback of this method is its complexity, and the time and effort involved. About 25 points must be recorded, in order to plot the hysteresis loop, and all points must be recorded under the same conditions (temperature, and so on), and the condition of maintaining the corresponding direction of motion of the input and output members for each of the hysteresis loop branches must be observed strictly.

The second of these methods is most frequently used. In those cases when higher dead zone accuracy measurement is required, the hysteresis loop method is used; this method results in a somewhat smaller dead zone magnitude.

Check of maximal output member velocity and velocity characteristic nonlinearity

In order to ensure constant maximal slide-valve displacement /113
when measuring the maximal velocity during output member travel, a force F_0 should be applied to the input member by means of a weight. Then, overcoming the action of the weight, the input member is displaced in the direction opposite the action of the weight, and the weight is abruptly released. The input and output members displace under the action of the weight with a maximal velocity which is determined with the aid of a contact place of known length and an electric stop watch which measures the time for the output member to travel the length of this plate. The contact plate should be located in the center of the output member travel, and its length should amount to 0.5 - 0.8 of the maximal output member travel, in order to exclude the influence of the transient processes with respect to velocity at the beginning and end of the travel.

The differences of the maximal velocities (or asymmetry) can be found from the formula

$$\Delta \bar{v} = \frac{2(v_{max} - v_{min})}{v_{max} + v_{min}} 100\%,$$

where v_{max} and v_{min} are the larger and smaller of the maximal retraction and extension velocities (absolute magnitudes).

In certain cases, it is necessary to determine the booster velocity characteristic (with respect to input member displacement) and its nonlinearity. This check is very complicated for boosters, since the booster is a closed system and it is difficult to obtain the output member motion regime with constant (nonmaximal) velocity. The

booster velocity characteristic can be obtained with the aid of a complex fixture, which is connected with the output and input members and makes it possible to maintain a constant established error as the booster input and output members move. In order to determine the velocity characteristic nonlinearity (see Figure 1.5), we identify on the characteristic a given velocity zone in which the nonlinearity is determined. The extreme points of the velocity characteristic obtained in the given zone are connected with a straight line, after which we find the maximal deviation Δv of the actual characteristic from this straight line. Then the velocity characteristic nonlinearity is /114

$$\Delta_{nonl} = \frac{\Delta v}{v_{max}},$$

where v_{max} is the maximal velocity in the given zone.

However, the accuracy of determination of the velocity characteristic and its nonlinearity by this technique is not high, particularly for boosters with complex kinematics. Therefore, it is better to evaluate booster velocity characteristic nonlinearity on the basis of the booster slide-valve flow characteristic, since the dependence of the input member and the slide-valve displacements is nearly linear, the hydraulic passage resistance is not large, and there is no leakage in the actuator cylinder piston. Sometimes it is necessary to determine the magnitude of the maximal booster velocity with an opposing load. This determination can be made with either an active or passive system for simulating the constant load. However, in this case, when establishing the given load, it is necessary to consider the actual nature of the load variation, shown in Figure 4.4.

Force developed and output member deflection under given load

When determining the maximal force which can be developed, one of the loading devices described in Section 4.2 is connected to the booster output member through a link with a dynamometer. Then a

weight is used to apply a force to the input member, and, as the output member moves, the opposing load is increased until the output member is brought to a complete stop in any intermediate position. The maximal force developed is determined from the dynamometer indication at the instant the output member comes to a stop.

The force developed can also be measured by another method. The output member is forced against the stop by an external force exceeding by 10 - 20% the nominal value of the force which can be developed. Then a force F_0 is applied to the input member by means of a weight in the direction opposing the action of the external force. By gradually reducing the external force, we can determine /115
from the dynamometer indication the booster force developed at the instant of output member initial movement. An active load simulation system is necessary when determining the force which can be developed using these techniques. We note that an incorrect technique is sometimes used to determine the developed force, in which, with the input member stationary, the force on the output member is increased until the output member begins to displace with minimal velocity. The force obtained in this way is actually the booster holding force, which exceeds the developed force by twice the magnitude of the booster piston friction.

The determination of the developable force can be made with the aid of a positional loading device (a spring, for example); in this case, the load corresponding to the output member extreme positions must exceed the expected developable force. In this case, a force F_0 is applied to the input member and, as a result, the output member displaces until it is stopped by the positional load. The dynamometer indication at the moment of stopping characterizes the magnitude of the developable force.

When determining the output member static deflection, the booster input member is fixed in any intermediate position. Then a specified force (for example, 90% of the developable force) is applied to the output member alternately in opposite directions, and

a dial gauge is used to determine the difference of the two output member positions. Half of this difference yields the magnitude of the output member static deflection. An active constant load simulation system is necessary when making this check.

Internal leakage and crossflow from system to system

When checking internal leakage, it is necessary to ensure minimal influence of slide-valve port obliteration on the measurement results. Leakage is usually determined in any intermediate position and at the two extreme input member positions. The input member should be moved through its travel two or three times prior to each measurement.

As was shown in Section 3.3, the instantaneous flow rate values /116
decrease as a function of time. Since commercial flowmeters which can indicate the instantaneous flow rate value are not available for small leakage flow rate magnitudes, the average leakage flow rate is usually determined by measuring the leakage volume ΔV_{le} during the time Δt: $Q_{le} = \Delta V_{le}/\Delta t$. It is obvious that the longer the time Δt, the smaller is the leakage flow rate magnitude measured in this way. Therefore, the leakage measurement time should be limited. It is recommended that after the booster has remained stationary for two to three seconds, the working fluid volume flowing from the return line be measured with the aid of a calibrated vessel in the course of one minute.

Determination of the working fluid leakage from each channel individually is very difficult for multichannel boosters. In this check, the working fluid is supplied under pressure to one of the channels, and the fluid level increase in a graduated tube connected with the fluid-filled internal chamber of the inoperative channel is checked. The basic difficulties in this determination are caused by spontaneous (in the absence of leakage) increase of the level in the graduated tube because of dissolved air evolution from the fluid

in the inoperative channel chamber, and heating of the fluid during the leakage measurement process. In order to ensure minimal temperature change, the working fluid should be maintained during the check at a temperature which does not differ by more than $\pm$ 5° C from the ambient air temperature. In order to fill the actuator working chambers, the output member should be operated for 5 - 10 minutes prior to making the check, after which the output member is set to one of the extreme positions and the hydraulic supply is deactivated. Since the degree of spontaneous level change in the graduated tube connected to the inoperative channel depends on the rate of air evolution from the fluid, it is recommended that gas at a pressure of about 10 N/cm^2 be fed into the top of the tube to reduce air evolution. An adapter must be provided on the tube to which a hand hydraulic press can be connected in order to recharge the chamber under test with fluid if necessary, and ensure the initial position of the fluid level in the tube required for the measurement when pressurization is applied. /117
The technique for determining the dynamic characteristics will be examined in Section 5.4.

5.3. Servocontrol Test Technique

The techniques for determining the basic parameters of servocontrols are basically analogous to the corresponding techniques for boosters, except for certain peculiarities. When checking servocontrols, it is necessary to consider the influence of electromechanical transformer magnetic hysteresis. The hysteresis loop width usually amounts to 3 - 6% of I_{max}. In order to reduce the hysteresis influence, the transformer should be demagnetized prior to measuring the dead zone and null shift. To accomplish this, we can perform in sequence two or three full throws with control signals 50 - 70% of I_{max}, 20 - 30% of I_{max}, and 8 - 10% of I_{max}, after which signals exceeding 10% of I_{max} are not permitted while making the measurements. Then without any time delay, the control current should be used to fix the actuator output member in any intermediate position and, by gradually varying the control current, we determine the

magnitudes and signs of the output member extension and retraction breakaway currents. The dead zone can be found from the algebraic difference of the breakaway currents. The null shift can be found from half the algebraic sum of the breakaway currents.

In certain cases, it is necessary to determine the servocontrol hysteresis loop width and the loop null shift. To do this we deflect the output member through two or three full cycles, using the maximal control signal, and leave the output member in one of the extreme positions. Then the current is gradually reduced, and the current at which piston rod motion begins is determined. Then this same technique is repeated with the output member on the other stop. From the measured breakaway current values, we determine the hysteresis loop width and the hysteresis loop null shift. The electromechanical transformer must also be demagnetized before determining the servocontrol velocity characteristic. The control currents for each of the specified points of the velocity characteristic must be established during the measurements with account for the null shift. Account for the null shift involves increasing the control current by the magnitude of the null shift with control signal polarity coinciding in sign with the null shift and, correspondingly, reducing the control current by the magnitude of the null shift with the opposite control current polarity. /118

These current values are used to plot the servocontrol velocity characteristic $v = v(I)$. The techniques for determining the velocity characteristic nonlinearity in a given control current range and the velocity characteristic asymmetry are analogous to the corresponding techniques for boosters. The techniques for determining the force developed, working fluid leakage, and cross flow from system to system (for multichannel units) are analogous to the corresponding techniques for boosters.

5.4. Technique for Determining Hydraulic Drive Dynamic Characteristics

Most often, the need arises for determining hydraulic drive frequency characteristics. The frequency characteristics are usually determined with an inertial load, and sometimes for given elasticity of the support and the connection between the output member and the load. When determining the frequency characteristics, a sinusoidal control signal (for the booster from a mechanical sine generator with electric motor or hydraulic motor drive, and for the servocontrol from a low-frequency periodic signal generator of the HGPK type) is applied to the input of the test hydraulic trive. The frequency characteristics are determined by analyzing the sinusoidally varying input and output signals, transformed into current or voltage. We determine sequentially at each specified frequency (for example, 1, 2, 5, 10, 15, and 20 Hz) the output and input signal amplitude ratio (in decibels) and phase shift (in degrees) for constant input signal amplitude.

For determining the input and output signal amplitudes, it is most convenient to use the DPV dual peak voltmeter, and for determining the phase shift — the NF low-frequency phasemeter.

Electronic frequency characteristic analyzers are sometimes /119
used to find the amplitude-phase characteristics.

However, in practice, because of better availability, galvanometer-type oscillographs are used to record the input and output signals on oscillograms. At the same time, a time marker signal having the frequency f_{tm} is recorded on the oscillogram. In order to plot the experimental hydraulic drive frequency characteristic, we analyze the set of oscillograms for various frequencies and constant specified input signal amplitude. The general form of the oscillogram is shown in Figure 5.1. We analyze the oscillograms to find the following quantities:

$A_{in\ i}$ and $A_{out\ i}$ — input and output signal amplitudes in mm at the frequency f_i;

T — input signal oscillation period in mm;

n_{tm} — number of timing marker oscillation periods per input signal oscillation period;

a_1 and a_2 — oscillogram segments which define the magnitude of the phase shift.

Then the actual recorded signal frequency is f_{tm}/n_{tm}, the gain in decibels is $Z = 20 \lg \frac{A_{out\ i}\, m_1}{A_{in\ i}}$ (m_1 is a scaling factor), and the phase shift in degrees is $\phi = \frac{360\ (a_1 + a_2)}{2T}$.

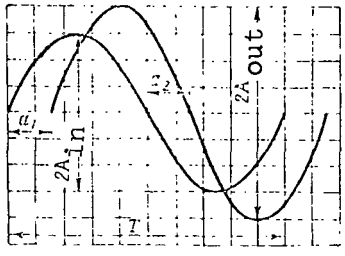

Figure 5.1. Typical oscillogram for determining the frequency characteristic

The servocontroller control current amplitude should be established using a milliammeter at a frequency no higher than 0.5 Hz, since at higher frequencies, the interia of the pointer-type instrument may distort the indications markedly. The selection of the oscillograph galvanometers should be made with care, since in certain cases they may distort the hydraulic drive frequency characteristics significantly. In order to minimize the galvanometer influence, they should both be of the same type, and have a passband no less than an order of magnitude higher than the maximal specified frequency. /120

Recording of the frequency characteristics of servocontrols, which are integrating elements, presents considerable difficulty, since at low frequencies the output member amplitude may reach excessively large values and the output member may reach the stops, while at high frequencies the amplitude is too low. Moreover, if there is even slight asymmetry of the control signal or velocity characteristic, the output member will "creep" from the initial position. In order to prevent output member "creep", the input electric

signal is tuned by introducing the required constant component from a low-frequency periodic signal generator.

In many cases, the servocontrol AFC and PFC are specified points in the form of the difference between the actual characteristics and the corresponding ideal integrating element characteristics. In this case, the phase shift difference can be found from the formula:

$$\Delta\varphi = \varphi - 90^{\circ}.$$

In this case, finding the AFC can be simplified, and we can avoid the laborious determination of the scaling coefficient m_1

$$\Delta Z = 20 \lg \frac{A_{out\ i}}{A_{in\ i}} \cdot \frac{A_{in\ 1}}{A_{out\ 1}} - A_i,$$

Where $A_{in\ 1}$ and $A_{out\ 1}$ are the input and output signal amplitudes in mm at a frequency of 1 Hz;

Z_i is the AFC of the integrating member, referred to a frequency of 1 Hz, and shown in Table 5.1.

TABLE 5.1.

f, Hz	1	2	5	10	15	20
Z_i, dB	0	6.05	14	20	23.5	26

In certain cases, and particularly for low output member amplitudes, the output signal shape differs markedly from the sinusoidal form because of the strong influence of hydraulic drive nonlinearities. In this case, we identify from the real periodic output signal recording curve, the first harmonic [32], and use this harmonic to determine the frequency characteristics. /121

In practice, it often becomes necessary to describe hydraulic drives analytically on the basis of experimentally determined frequency characteristics. The possibility of quite accurate linear approximation of hydraulic drive dynamic characteristics can be evaluated on the basis of frequency characteristic agreement for different control signal amplitudes (in the operating range), and on the basis of output signal shape similarity to the ideal sinusoid. The hydraulic gain for given control signal amplitude can be determined with the aid of the logarithmic AFC from the ordinate of the zero-slope segment at low frequencies for hydraulic drives in which the output signal is the output member displacement (booster, for example), or from the ordinate of the low-frequency rectilinear segment for ω = 1 rad/sec, for hydraulic drives in which the output signal is the output member velocity (servocontrols, for example). The techniques for finding dynamic models from experimental AFC and PFC of various systems were examined in [9].

When experimental determination of hydraulic drive transient processes with step-wise change of the input signal is required, we record on a light-beam oscillograph the input and output signals, and also the timing marker signal. In order to find the servocontrol lag time τ_{sc} and velocity rise time constant T_{sc}, it is necessary to record on the oscillogram the signal from the output member velocity sensor, which provides considerably higher accuracy of τ_{sc} and T_{sc} determination than the signal from the output member displacement sensor. The quantity τ_{sc} is defined by the time between the initiation of control signal application and the initiation of output member velocity change (see Figure 1.11b). The time constant T_{sc} can be found from the time when the output member velocity signal /122
reaches 63% of the steady-state value corresponding to the control signal.

We note that the phase shift caused by T_{sc} and τ_{sc} usually exceeds the phase shift obtained when determining the servocontrol frequency characteristic. This is due to the fact that the lag

determined in the transient process includes both time and coordinate lags [32]. The experimental AFC is influenced only by time lag, while the coordinate lag causes distortion of the output signal sinusoidal curve shape.

When determining booster dynamic stiffness, the input member is rigidly connected with the test stand frame, and a harmonically varying force is applied to the output member. The source of this force may be an electric motor or hydraulic motor with variable rotational speed, connected by means of an adjustable eccentric and flexible linkage with the output member. The required force amplitude, determined with the aid of a tensometric force sensor, is established in turn at each frequency by regulating motor speed and eccentricity. The output member force and displacement signals, whose amplitude ratio defines the booster dynamic stiffness, are recorded by a light-beam oscillograph on an oscillogram. Usually, with frequency increase to 10 - 15 Hz, the dynamic stiffness increases somewhat in comparison with the static stiffness and then remains practically constant with further frequency increase. The gradual stiffness increase which occurs in the low-frequency region is explained basically by reduction of the distribution slide-valve opening by the feedback as the force application frequency increases.

Verification of booster dynamic stability must be carried out on a stand with simulation of the output member load and, also, mounting stiffness and elasticity of the connection between the output member and the load. In order to verify the absence of self-maintained oscillations and vibrations, it is advisable to perform sharp reversals of the input member and monitor the transient processes at the output member. In this case, undamped oscillations at the output member should not take place, and the number of overshoots in the transient process should not exceed four or five. In /123
order to verify the output member travel smoothness, it is convenient to use low-frequency (0.02 - 0.05 Hz) harmonic displacement of the input member with amplitude at which output member velocity equal to 20 - 30% of the maximal velocity is reached.

In determining hydraulic drive dynamic characteristics, precise simulation of the required forms of output member loading [factors b_3 in (2.1)] is very important. We must make certain that the magnitude of the additional loads, not specified in the technical documentation and due to the characteristics of the hydraulic drive loading system are minimal. For example, when determining the dynamic characteristics of boosters with inertial and positional loads, the influence of the friction force due to the inertial load simulation system may distort the results markedly. Still larger distortions of the results may take place when determining the frequency characteristics of boosters and servocontrols with positional load, because of the "parasitic" inertial load influence. Therefore, it is recommended that prior to making dynamic tests of hydraulic drives, in addition to static calibration of the loading system, an oscillographic study be made of the input member load variation for the typical operating regimes.

The following measurement accuracies of the basic controllable hydraulic drive external factors and output parameters are recommended when making hydraulic drive tests:

supply and return pressure . . . . . $\pm$ 1%;
working fluid flow rate . . . . . $\pm$ 2%;
force developed . . . . . $\pm$ 2%;
output member load (with account for variation as function of output member travel) . . . . . $\pm$ 10 %.

The questions associated with measurement errors, average value estimates, and measurement result analysis were examined in detail by Rumshinskiy in [38].

CHAPTER VI

MECHANICAL INPUT TESTS OF HYDRAULIC DRIVES

6.1. Influence of Mechanical Inputs on Hydraulic Drives

Mechanical inputs cause a considerable fraction of aircraft on- /124
board equipment failures. According to the data presented in [15], vibration and shock led to 41% of the failures in a groups of 2600 items of airplane hydraulic and pneumatic equipment.

Vibrations are the most hazardous form of mechanical input. Since the random vibrations which occur in flight are usually simulated in bench vibration tests by harmonic vibrations, it is advisable to examine the influence of both random and harmonic vibrations on hydraulic drives.

Vibrations may influence the stability of certain types of hydraulic drives. The greatest danger is that presented by the action of vibrations of the velocity characteristic of servocontrols in the presence of resonance of the first amplification stage of the hydraulic amplifier in the frequency range being investigated. In this case, for the considered type of servocontrol (see Figure 19), under the action of vibrations with frequency close to the resonant frequency, forced mechanical vibrations of the slide valve with considerable amplitude develop. These high-frequency oscillations,

acting like intense noise superposed on the control signal [32], reduce the output member velocity. The degree of velocity reduction increases with increase of the oscillatory tendency of the slide-valve, approach of the vibration frequency to the resonant frequency, and with increase of the amplitude of the vibrational acceleration acting in the distribution valve area. Vibrations acting in the /125 direction of the servocontrol hydraulic amplifier nozzle axis present the greatest danger.

For servocontrols with potentiometer feedback sensors, vibrations may also lead to contact disruption in the potentiometer at frequencies close to the natural frequency of the current pickoff. We note that harmonic vibrations with constant frequency equal to the resonant frequency of the corresponding servocontrol structural component have the greatest influence on hydraulic drive characteristics. Under the action of vibrations with sweeping frequency (or random vibrations), the characteristics change to a much smaller degree, and in many cases do not change at all.

Vibrations have practically no influence on the characteristics of boosters, which are comparatively simple hydraulic power units.

Vibrations may have a significant influence on hydraulic drive strength. Deformations of hydraulic drive structural components take place under the action of vibrations on the hydraulic drive mounts because of the occurrence of inertial forces. In this case, alternating stresses arise which may lead to fatigue failure of the hydraulic drive structural elements in which the stresses are maximal. Resonant phenomena in structural elements with low damping and natural frequency which falls in the operating vibration frequency range present the greatest danger from the viewpoint of the possibility of fatigue failure from the influence of vibrations. A characteristic of hydraulic drives, which are massive power systems, is the fact that the natural frequencies of their structural elements (usually above 200 Hz) exceed by several times the natural frequency of the aircraft hydraulic drive mounting units (usually below 80 Hz).

Therefore, the hydraulic drive mounting usually has a damping influence on the drives, and attenuates the resonant phenomena influence.

In the case of mechanical inputs, there may be fatigue failures of the hydraulic drive structural elements, with the failure mode depending basically on the loading in the process of fatigue crack initiation and growth, on the strength of the structural element, /126
determined by its geometric shape, material properties, and surface condition, and also on the external medium (temperature, corrosivity, and so on). Most often we encounter typical fatigue fractures, which take place with a number of loading cycles, exceeding hundreds of thousands, and low-cycle fractures, which occur after a comparatively small number of static loading cycles (usually no more than 10^4 [16].

The typical fatigue curve of a steel specimen (Figure 6.1) is characterized by the following parameters:

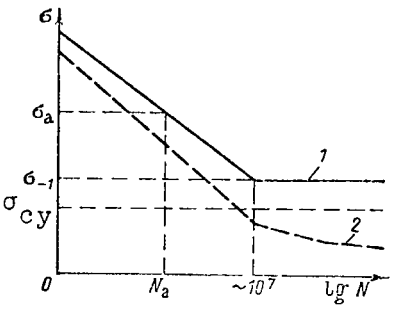

Figure 6.1. Typical fatigue curves of steel (1) and nonferrous metal (2) specimens

— the endurance (fatigue) limit σ_{-1} — the largest value of the maximal cycle stress with the action of which fatigue failure of the specimen does not occur after an arbitrarily large number of cycles (usually σ_{-1} is reached when operating for a number of cycles greater than 10^7;

— limited endurance limit σ_a — the largest value of the cycle stress amplitude with the action of which fatigue failure does not occur for a definite (specified) number of cycles;

— the test base — the specified number N_a of stress cycles;

— fatigue life — the number of cycles withstood by the specimen prior to failure at a definite (specified) stress.

Ivanova [16] introduced the concept of cyclic elastic limit σ_{cy}, which is less than σ_{-1}. The quantity σ_{cy} characterizes the maximal stress for which irreversible crystal lattice distortions, which are the beginning of the fatigue failure process, do not show up for any number of loading cycles. Cyclic stresses less than σ_{cy} do not have any practical influence on the material of the parts. Cyclic /127
stresses higher than σ_{cy} by less than σ_{-1} cannot lead to fatigue failure of the part; however, the crystal lattice irreversible distortion process begins at these stresses.

The dependence of fatigue life N_i on the stress which causes failure is described for steel specimens by the Oding-Weibull equation:

$$N_i = k_{\sigma_1} (\sigma_i - \sigma_{-1})^{-m_1},$$

where k_{σ_1} and m_1 are constants which depend on the material.

For the limited fatigue strength segment of the steel specimens, it is more convenient to use the Baskvin equation [37]:

$$\sigma_i = k_\sigma N_i^{-m}, \qquad (6.1)$$

where k_σ and m are constants which depend on the material.

The Baskvin equation is also valid for the nonferrous alloys, including the aluminum alloys, for which the endurance limits are close to zero. The right branches of the fatigue curves for these alloys approach the abscissa axis asymptotically (curve 2 in Figure 6.1).

We note that temperatures up to + 200° C for steel parts, and up to + 120° C for alluminum alloy parts, have practically no influence on the fatigue strength, which permits, in most cases, conducting vibration strength tests at normal temperature.

During the action of vibrations on hydraulic drives, fatigue damage accumulation takes place in their structural elements. The most widely used fatigue damage accumulation theory is that proposed by Palmgren and Mainer, based on a linear accumulation law [40, 45]. In accordance with this theory, fatigue damage accumulation is characterized by the damage index D:

$$D=\sum_{i=1}^{k}\frac{n_i}{N_i}, \tag{6.2}$$

where n_i is the number of stress cycles at the frequency f_i with stress amplitude σ_i; N_i is the number of cycles to fatigue failure with the constant stress amplitude σ_i. /128

The damage index D is a dimensionless number which varies from zero to one. For a linear fatigue damage accumulation law, failure should occur when the index D reaches one. However, as was shown in [45], in the case of successive loadings with different constant stress amplitudes, the linear law is valid only in the middle zone of the fatigue curve. However, when the stress amplitude decreases from large to small values during the tests, extreme damage takes place in the early cycles, and failure occurs for values of D considerably less than one. Conversely, when the stress amplitude increases from small to large values, hardening of the material takes place in the early cycles, and failure occurs for values of D considerably larger than one. However, in spite of this drawback, the linear law is widely used in engineering practice because of its simplicity, with stresses exceeding $0.8\,\sigma_{-1}$ [45] being taken into account.

In the random vibration case, there is no definite order of succession of low and high stress amplitudes; therefore, the hardening effect of low stresses and the destructive effect of high stresses compensate for one another, and the index D is close to one at failure [40].

The highest stresses in hydraulic drive structural elements arise under the action of harmonic vibrations with frequency equal to the resonant frequency of the element. In the case of vibrations with sweeping frequency (in the resonant zone), for the same vibrational accelerations, the maximal stresses decrease since the resonant phenomena cannot develop fully. In this case, the higher the frequency sweep rate, the smaller the magnitude of the maximal stresses.

One characteristic of hydraulic drives is their variable rigidity and, accordingly, variable natural frequency, which depends on hydraulic drive output member position. Another characteristic of certain types of hydraulic drives (primarily hydraulic motor drives) /129 is the fact that they are themselves powerful sources of vibrations whose intensity in many cases exceeds that of the external vibrations. These characteristics of hydraulic drives must be considered in developing the vibration test technique.

In addition to hydraulic drive structural element failure, vibrations may also lead to unintentional loosening of screws, nuts, and so on, relaxation of the tension of threaded connections, disruption of adjustment made with the aid of threaded connections, failure of electric component soldered terminals, breaking of electrical component terminals, abrasion of wiring cables on the surface of parts with which they come into contact, and so on.

The action of shock pulses on hydraulic drives may lead to the development of significant stresses in the structural elements, and these stresses may cause low-cycle fracture of these elements. The maximal stresses which arise under the action of shock pulses depend on shock pulse shape, shock acceleration magnitude, and also on the relationship between shock pulse duration τ_{sh} and hydraulic drive structural element natural vibration period T_i. If for a shock pulse of sinusoidal form $\tau_{sh} > (2 - 3)\, T_i$, then the same stresses σ^*_{sh} arise as under the action on the hydraulic drive of a constant

linear acceleration equal to the shock acceleration. For $\tau_{sh} \leq$ (0.1 - 0.3) T_i, the shock pulse cannot excite the hydraulic drive structural elements, and stresses less than σ^*_{sh} arise under the action of this pulse. The highest stresses, exceeding σ^*_{sh}, usually arise under the action of a shock pulse with τ_{sh} = (0.5 - 1) T_i [18]. Since hydraulic drive structural element natural frequencies usually exceed 200 Hz, under the action of shock pulses with duration greater than 10 - 15 ms, maximal stresses σ^*_{sh} arise which are independent of shock pulse duration. The action of shock accelerations has practically no influence on hydraulic drive characteristics in view of the short duration of shock pulse action. We note that the action of vibrational accelerations and shock accelerations on hydraulic drives /130 leads to fatigue damage accumulation in the same structural elements; therefore, hydraulic drive tests for these input modes must be carried out on the same assembly.

Linear accelerations may have a marked influence on servocontrol characteristics, leading to increased null shift. Most critical for the servocontroller are linear accelerations acting in the direction of the hydraulic amplifier nozzle axis. The inertial forces which arise under the action of linear accelerations cause shift of the hydraulic amplifier flapper and slide-valve, and, for the scheme shown in Figure 1.9, these shifts frequently compensate one another and the overall null shift usually does not exceed 0.1% of I_{max} under the action of a linear acceleration equal to 1 g. The influence of linear accelerations acting perpendicular to the slide valve and piston axes may cause increase of the friction force on these components; however, this friction increase is negligibly small.

Linear accelerations have practically no influence on booster characteristics. Linear accelerations usually have only a slight influence on the strength characteristics of hydraulic drives, since the maximal stresses in the structural elements from the action of linear accelerations do not exceed the maximal stresses from the

action of vibrational or shock accelerations, and the number of linear acceleration loading cycles is comparatively small.

The effect on hydraulic drives of accelerations which arise during shipping is similar to the action of vibrational and shock accelerations.

In contrast with vibrations, acoustic noise acts on the entire surface of the hydraulic unit. However, because of the relatively small surface area and the large mass of the hydraulic drives, they can be considered insensitive to acoustic noise.

Thus, for aircraft hydraulic drives, we can recommend conducting the following forms of mechanical input tests:

— tests for vibration strength;

— tests for vibration stability (only for hydraulic drives of the servocontroller type);

— tests for shock strength;

— tests for resistance to the action of linear accelerations /131 (one-time tests for servocontrollers only).

6.2. Mechanical Test Equipment

Conventional shake tables which provide the required frequencies and accelerations (amplitudes) are used for hydraulic drive vibration tests. Single-component mechanical or electrodynamic shake tables which develop harmonic vibrations are most frequently used.

In view of the considerable mass of the hydraulic drives, special attention should be devoted to design of the brackets for the vibration tests. The brackets must provide hydraulic drive

stiffness which is close to the operational stiffness, since the use of stiffer brackets may lead to more marked manifestation of the resonant phenomena in hydraulic drive vibration tests than during flight operation. The brackets should ensure minimal lateral accelerations in the hydraulic drive attachment area.

Since hydraulic drives are comparatively large items with elements having high natural frequencies, it is important in hydraulic drive vibration testing to minimize the nonuniformity of the vibration acceleration distribution over the table surface, and minimize the multiharmonic vibration mode distortions. Particularly critical in this regard are the mechanical shake tables (for example, VUS 500/200), for which, in many cases, when operating in the low-frequency regimes, high-frequency components arise with acceleration exceeding the required value by several fold.

Equipment for measuring vibration parameters has been examined in [17, 18, 25]. Most frequently, accelerometers of the piezoelectric and induction types are used to measure the vibrational accelerations. When using mechanical shake tables, prior to conducting the vibration tests, an oscillographic study of the signal from the high-frequency piezoelectric accelerometers must be made to verify the absence of any unacceptable high-frequency vibration component.

To conduct hydraulic drive tests for shock strength, we usually /132
use shock stands which create shock accelerations in the vertical direction upon impact on elastic anvils of a freely falling table with the unit under test mounted thereon. The most frequently used shock pulse, having the form of a sinusoid half-period, is characterized by the maximal pulse acceleration and duration.

The pulse duration is determined by the width of the base of the first shock pulse overshoot. The influence of the second and subsequent shock acceleration oscillations, due to table rebounds, can be neglected because of their smallness. The required shock acceleration magnitude and shock pulse duration are regulated by

varying the table fall height and selecting the interchangeable soft and hard rubber elastic pads. The maximal shock acceleration magnitude can be determined with the aid of either a calibrated piezoelectric accelerometer and an electric oscillograph, or a peak acceleration sensor. The pulse duration is determined from the piezoelectric accelerometer signal by photographing the shock pulse on the screen of an electronic oscillograph or light-beam oscillograph. The number of shocks per minute is regulated by varying the rotation frequency of the crank controlling the table motion. Usually, shock strength tests are conducted at a frequency of from 40 to 80 impacts per minute.

The shock stands are equipped with a shock counter to permit performing a specified number of shock pulses during the tests.

For conducting hydraulic drive tests under linear accelerations, use is made of centrifuges which, as they rotate in the horizontal plane, create radially directed centrifugal (linear) accelerations [8]. When testing servocontrollers, a hydraulic reservoir is required on the centrifuge to provide hydraulic supply, and a slip-ring system — to supply the controlling electric signal to the unit under test and pick off the signal from the feedback sensor. A counterweight is mounted opposite the test unit to dynamically balance the loaded centrifuge.

In order to test for the effect of accelerations which arise /133
during shipping, use is made of either conventional shake tables or shock stands or special setups which simulate shipping acclerations in the bodies of trucks traveling over dirt, gravel, or asphalt roads at various speeds [8].

When conducting mechanical input tests, hydraulic supply is provided to the hydraulic drives from the hydraulic test stand through flexible hoses, and electric power is provided through cables.

6.3. Techniques for Conducting Hydraulic Drive Mechanical Tests

Tests of hydraulic drives with mechanical inputs are usually conducted sequentially along three axes under normal climatic conditions without load on the output member. Hydraulic drives are tested for vibrational strength and shock strength with hydraulic supply connected and with the operational position of the piston rod (or with the extended piston rod position, which is worst from the strength viewpoint). The mounting of the test hydraulic drive to the bracket must be the same as under operating conditions.

The most complex form of mechanical input tests are the vibrational tests. At the present time, hydraulic drive testing under the action of harmonic vibrations is most widely used. These vibrations are characterized by the displacement amplitude s in millimeters, vibrational load factor I_g in g units, and vibration frequency f in Hertz, where

$$I_g = \frac{sf^2}{250}.$$

When conducting vibration hydraulic drive tests, it is necessary to ensure obtaining the specified vibrational accelerations (or amplitudes) at the points of attachment to the mounting bracket. Determination of the vibrational accelerations using a pickup installed on the vibrational table can lead either to the development of excessive vibrational loads on the test hydraulic drive because of bracketry resonance, or to underloading of the hydraulic drive. When mounting the hydraulic drive on two hinged supports (stationary /134
on the housing and moving on the piston rod), the required vibrational acceleration should be specified on the stationary support.

The vibrations of hydraulic drives in the brackets have a three-dimensional nature, even with unidirectional table vibration. Therefore, during the vibration tests, we must also monitor the lateral accelerations at the point of hydraulic drive attachment, which in

certain cases may reach the magnitudes of the driving vibrational accelerations. In this case, we consider these tests to be qualifying tests with respect to two (three) axes.

In most cases, the vibration tests start with determination of the resonances in a specified frequency range. The resonant frequency is considered to be that at which we observe either determination of the hydraulic drive characteristics or mechanical resonance of the structural elements. Only mechanical resonance is monitored for the hydromechanical drives. One of the two following techniques can be used to determine mechanical resonance. In accordance with the first technique, under the action of harmonic vibration with constant driving vibrational acceleration amplitude, the vibration frequency is varied smoothly (at a rate of one octave per minute), and the ratio of the vibration accelerations in the areas of the hydraulic drive elements being monitored to the driving vibrational acceleration on the support is recorded. In this case, we monitor the vibrational accelerations in the direction of the driving axis for those elements for which the expected resonant frequencies are lowest (basically for cantilever elements). Usually, the vibration frequencies for which these vibrational acceleration ratios are minimal and exceed two are considered to be resonant frequencies.

Since the primary danger for hydraulic drives subject to vibrations lies in the maximal stresses in the structural elements, the determination of the resonant frequencies with the use of strain gauges bonded in the hydraulic drive areas which are critical in regard to vibration frequencies for which the stresses are maximal is the best approach. We note that in many cases, the resonant frequencies determined from the vibrational acceleration ratio exceed somewhat the resonant frequencies determined from the stresses. This is due to the fact that only those vibrational acceleration components which cause deformation of the hydraulic drive structural /135
elements affect the stresses; the accelerations associated with oscillations of the hydraulic drive as a solid body do not cause stresses in the structural elements. However, determination of resonance on the basis of the stresses is a technically complex operation, and can be used only in the research study stage.

Since the moving system, consisting of the massive hydraulic drive being tested, bracketry, and the vibration table is a complex multimass three-dimensional system, the appearance of "false" resonances due to the influence of the bracketry and the vibration table may appear when seeking hydraulic unit resonance.

In many cases, more exact results can be obtained with the aid of the second technique for determining the resonant frequencies of hydraulic drive structural elements. In accordance with this technique, the hydraulic unit is suspended on a cable, and natural vibrations are excited in its structural elements (for example, by striking with a rubber-covered hammer). The natural frequencies of the structural elements being investigated are determined from oscillograms of the transient processes, recorded with the aid of vibration pickups installed on these elements.

The basic general techniques used in hydraulic drive tests for vibrational stability and vibrational strength are the methods of random vibrations (wide-band or narrow-band) and harmonic vibrations (sweep-frequency, fixed and resonant frequencies). At the present time, most Soviet and foreign standards (for example, GOST 16962-71, MIL-STD-810B) provide for use of the sweep frequency method as the basic vibration testing technique.

Tests for vibrational stability should be conducted only for hydraulic drives of the servocontroller type. When testing using the sweep-frequency technique, the specified frequency range is divided into several (5 - 8) subranges, and the vibration frequency is varied smoothly in each subrange at constant amplitude (or acceleration) and rate of about one octave per minute in the course of the time required for measuring the dead zone, null shift, output member velocity, and checking feedback potentiometer contact. /136

The tests of hydraulic drives for vibrational strength by the sweep-frequency method are made similarly in the course of the time required to run through no less than 10^7 cycles (usually 50 - 70

hours). This number of cycles is selected on the basis of the fatigue strength criterion. In certain cases, in the presence of resonances, provision is made for additional running for a given number of cycles (10^6, for example) with frequency sweep in the zone near the resonant frequency (for example, in the zone $\pm$ 20% from the resonant frequency) at a rate of about 10 octaves per minute.

Sometimes, when only mechanical vibration stands are available, the fixed-frequency method is used, with additional operation at the resonant frequencies for a considerable number of cycles (10^6, for example). In this vibration testing method, the hydraulic drive experiences loads which are tens of times more severe than under operating conditions; therefore, the use of this technique is not advisable. However, if it becomes necessary to use this technique, the duration of the operation at the resonant frequencies should not exceed 10^4 cycles.

We note that, although the sweep-frequency method makes it possible to simulate more exactly the action of operational vibrations than does the fixed-frequency technique, the former method permits only very approximate reproduction of the actual vibrational loading pattern, since aircraft vibrations are random. Therefore, the random vibration test technique is beginning to be used more widely at the present time.

Random vibrations are reproduced either by recording on magnetic tape the typical operating vibrational loads, or an artificial signal created by a white noise generator with corresponding set of filters. There is no doubt that in the future this test technique will find broad application, and will become mandatory in testing complex aircraft equipment items.

Testing of hydraulic drives for vibration strength is a quite lengthy process; therefore, it is advisable to examine the question of accelerating these tests. As the basis for the accelerated vibration strength testing method, we can take constancy of the fatigue /137

damage accumulation magnitudes in the normal and accelerated vibration tests, i.e., constancy of the damage index D [see (6.2)]. In tests using the harmonic vibration technique, increase of the vibration accelerations from I_g to I_{ga} is usually used to accelerate the tests, in this case in accordance with (6.1) and (6.2) , the number of loading cycles (and duration of the vibration tests) decreases in each frequency subrange (or at each fixed frequency) by a factor of $(I_{ga}/I_g)^m$. The quantity m can be taken equal to 5 (this value is taken, for example, in the British Standard for Aircraft Equipment BS-3G100).

In random vibration testing acceleration of the tests is achieved by increasing the spectral density level; in this case, the coefficient m should be taken equal to 2.5.

A second way to accelerate vibration tests is to increase the running time at the resonant frequencies. For example, in the U.S.A. standard MIL-STD-810B, the total vibration test duration is nine hours, and cycles are run for 30 minutes at each resonant frequency. If the number of resonant frequencies in each axis exceeds four, 30 minute periods are run for the four highest resonances. The remaining time is spent generating vibrations by the sweep-frequency method in the given frequency range (from 5 to 500 Hz, or from 5 to 2000 Hz).

When testing hydraulic drives for shock strength, a weight equal to that of the test hydraulic drive and bracketry is used to adjust the required shock pulse on the shock stand. Then the hydraulic drive and its bracket are mounted on the stand and subjected to the specificed number of pulses alternately along the three coordinate axes. The tests should be conducted with the hydraulic supply and electric power connected.

If it becomes necessary to accelerate the shock strength tests, the shock acceleration can be increased while maintaining the pulse

duration; in this case, the equivalent number of shocks decreases by a factor of $(I_{ga}/I_g)^m$, where m = 5. We note that when accelerating tests for shock strength and vibration strength, it is necessary that the maximal stresses in the hydraulic drive structural elements not exceed the yield point in the accelerated regime with the acceleration I_{ga}. /138

It is recommended that testing of servocontrollers for resistance to the action of linear accelerations be conducted only under linear accelerations in the direction of the slide valve and nozzle axis. The relationship between the linear acceleration I_g in g units, centrifuge rotational velocity n_{ce} in rpm, and test hydraulic drive mounting radius r_{ce} in cm, has the form

$$n_{ce} = \frac{30}{\pi}\sqrt{\frac{I_g g}{r_{ce}}} \approx 300\sqrt{\frac{I_g}{r_{ce}}}. \qquad (6.3)$$

Creation of the required linear acceleration I_g is achieved by varying r_{ce} and n_{ce} [see (6.3)]. For hydraulic drives, the radius r_{ce} is determined on the basis of the distance from the centrifuge platform center of rotation to the center of gravity or the hydraulic drive element, which is most sensitive to the action of linear accelerations (for example, to the servocontroller distribution valve).

In tests of servocontrollers for resistance to the action of linear accelerations, we usually check the dead band, null shift, and maximal output member velocity. The checks should be made on the servocontroller mounted on the platform prior to, during, and after the tests. Prior to the tests, it is necessary to make certain that there is no loss of pressure in the centrifuge hydraulic reservoir at the maximal working fluid flow rate, and that the monitored supply pressure ahead of the reservoir and return pressure are practically equal to the pressures immediately at the hydraulic unit fittings.

When conducting tests with mechanical inputs, it is recommended that the following parameter measurement accuracy be provided:

vibration frequency:
  up to 50 Hz . . . . . ± 2 Hz;
  above 50 Hz . . . . . ± 5%;
vibration displacement amplitude. . ± 20%;
acceleration . . . . . ± 20%;
resonant frequency . . . . . ± 1 Hz.

CHAPTER VII

HYDRAULIC DRIVE CLIMATIC TESTS

7.1. Influence of Climatic Inputs on Hydraulic Drive Characteristics

High and low working fluid temperatures have the strongest influence on hydraulic drive characteristics. Since slow variation of the working fluid and ambient temperatures is characteristic for aircraft hydraulic drives, the hydraulic drive parameters are specified and verified under steady-state thermal regimes. /139

We shall examine the relationship of working fluid and ambient air temperature influence on hydraulic drive thermal regime*. We can take as the thermal regime characteristic parameter the temperature on the external surface of the unit wall, which is a function of the working fluid and surrounding medium parameters, structural parameters, and operating regime of the unit. We examine the case of heat exchange between the unit and a warmer ambient medium (the temperature pattern is similar in the reverse case). The following basic assumptions are made in calculating the thermal regime of the unit.

*Calculations were made by Yu. V. Tochilin.

1. The influence of thermal radiation is not considered.

2. The heat transfer calculation is made using the formulas for flat walls, since hydraulic drives can be considered thin-wall shells of various shapes.

3. The temperature field of the unit is considered to be steady-state, i.e., the working fluid and surrounding medium temperatures do not change in time, but only along the surface of the unit.

Thus, in the hydraulic drive thermal regime model examined, /140
heat transfer from the surrounding medium to the fluid in the unit takes place through the separating wall by means of convection and conduction, and, in this case, convection is observed only in liquids and gases, and represent heat transport by displacing fluid particles. We can take as the basic relations for convective heat transfer beween the hydraulic drive and ambient medium, the Newton-Riemann and Bio-Fourier laws [26]:

$$q=\alpha\Delta t \tag{7.1}$$

and

$$q=-\lambda\ \mathrm{grad}\,t, \tag{7.2}$$

where q — is the specific thermal flux; λ is the wall thermal conductivity; t is temperature; α is the heat transfer coefficient.

Using (7.1) and (7.2), we can obtain the basic heat transfer equations for the individual segments of the computational model adopted. The specific thermal flux from the air boundary layer to the outer surface of the wall

$$q=\alpha_{\mathrm{a}}(t_{\mathrm{a}}-t_{w}),$$

the specific thermal flux from the outer surface of the wall to the inner surface

$$q=\frac{\lambda}{\delta}(t_{w}-t_{w1})$$

(here, δ is the wall thickness), and the specific thermal flux from the inner surface of the wall to the working fluid

$$q = \alpha_{wf}(t_{w1} - t_{wf}).$$

The subscripts "a", "wf", "w", and "w1" refer to the air, working fluid, outer and inner surfaces of the hydraulic drive wall.

Equating these specific thermal flux values and solving the resulting system of equations for the characteristic temperature t_w, defining the assembly thermal regime, we obtain (7.3)

$$t_w = \frac{t_a \alpha_a \left(\alpha_{wf} + \frac{\lambda}{\delta}\right) + \frac{\lambda}{\delta} \alpha_{wf} t_{wf}}{\alpha_{wf}\left(\frac{\lambda}{\delta} + \alpha_a\right) + \frac{\lambda}{\delta}\alpha_a}. \qquad (7.3)$$

The heat transfer coefficients α_a and α_{wf}, appearing in (7.3) /141 are functions of the surrounding medium and working fluid parameters:

$$\alpha_a = (4{,}8 + 3{,}4 w_a)\sqrt{\frac{p}{2{,}6 t_a}}, \qquad (7.4)$$

$$\alpha_{wf} = \varepsilon \frac{\mathrm{Nu}\,\lambda_{wf}}{D}, \qquad (7.5)$$

where w_a is the surrounding air velocity in m/s; p is the surrounding air pressure in mm Hg (133 N/m^2); $\varepsilon = \varepsilon(l/D)$ is a correction factor; l, D are characteristic dimensions of the unit; Nu is the Nusselt similarity criterion, characterizing heat transfer in the heated fluid layer. For the turbulent working fluid flow regime, Mikheev [26] obtained the following expression for Nu:

$$\mathrm{Nu} = 0{,}021\,\mathrm{Re}^{0{,}8}\,\mathrm{Pr}^{0{,}43}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_{st}}\right)^{0{,}25},$$

where $\mathrm{Re} = \frac{v_f D}{\nu}$ is the Reynolds number; $\mathrm{Pr} = \frac{\nu}{a}$ is the Prandtl number; v_f is the fluid velocity in m/s; ν is the fluid kinematic viscosity in m^2/s; a is the fluid thermal diffusivity in m^2/s.

Analyzing the relations (7.4) and (7.5), we find that for typical hydraulic drive parameters, the wall heat transfer coefficient exceeds by about 100 - 1000 times the heat transfer coefficient of the surrounding air. This relationship of the air and fluid heat transfer coefficients leads to a situation in which the temperature at the outer surface of the hydraulic drive wall is determined basically by the temperature of the working fluid, and is nearly independent of the surrounding air temperature. Results of t_w calculation for typical medium-power booster parameters are shown in Figure 7.1. We see from these results that, for constant working fluid temperature + 200° C and with variation of the surrounding medium temperature from 0 - 250° C, the temperature at the outer surface of the wall varies from 193 to 201.5° C, i.e., only by 8.5° C. The experimental results confirm the validity of the calculated relations. We see from the results of temperature measurement for a

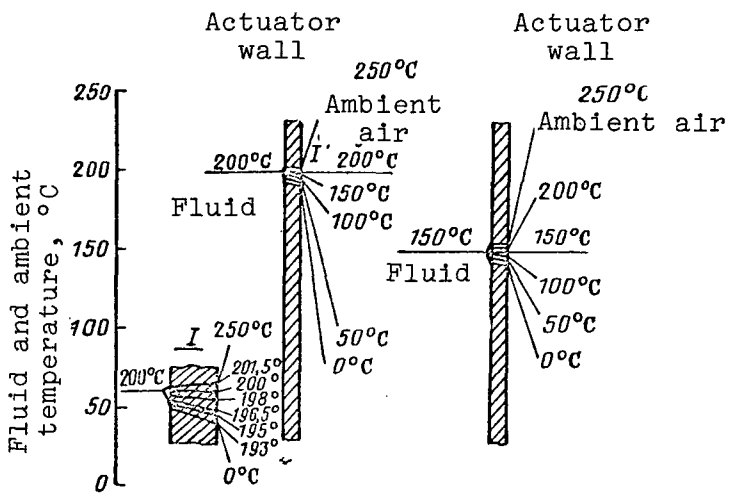

Figure 7.1. Calculated dependence of hydraulic drive power cylinder outer wall surface temperature on surrounding medium temperature

medium power booster, shown in Figure 7.2, that for constant air /142
temperature, equal to + 5° C, and gradual variation of the working fluid temperature from + 20° C to the steady-state temperature + 160° C, the temperature at the outer surface of the booster power cylinder was 5 - 8° C less than t_{wf}, and the temperature at the

the outer surface of the distributor valve body was 2 - 5° C less than t_{wf}.

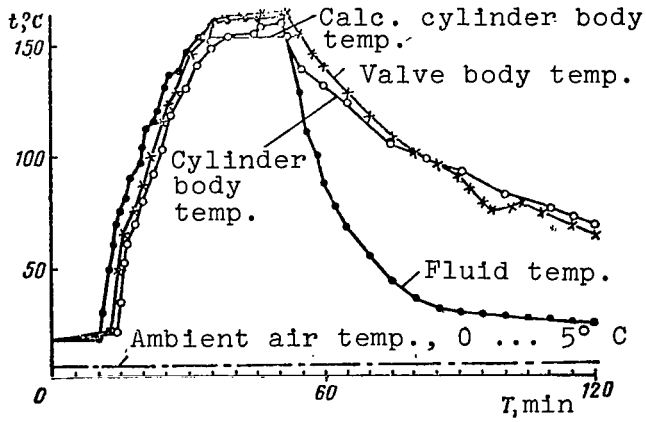

Figure 7.2. Experimental curves of booster body temperature as function of thermal regime

We note that because of the presence of powerful thermal bridges, all the hydraulic drive elements take essentially the working fluid temperature. Figure 7.3 shows the variation of servocontroller electromechanical transformer winding resistance as a function of working fluid and ambient temperatures, from which we see that the temperature in the electromechanical transformer area is practically independent of the ambient temperature. Figure 7.4 shows the results of temperature measurement in the servocontroller feedback potentiometer chamber, which for t_{wf} = +160° C and t_a = + 20° C was only 5 = 7° less than t_{wf}.

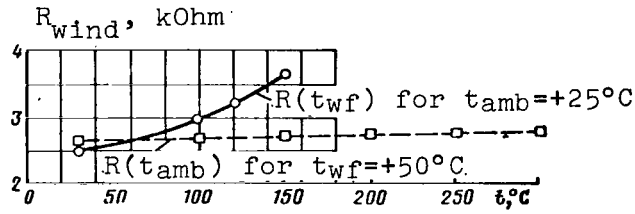

Figure 7.3. Experimental variation of servocontroller electromechanical transformer winding resistance with working fluid and ambient temperatures

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Since the hydraulic drive thermal regime is determined by working fluid temperature, the hydraulic drive characteristics are also determined by working fluid temperature and are practically independent of ambient temperature. This is confirmed by the test results of an experimental servocontroller, shown in Figure 7.5. /145

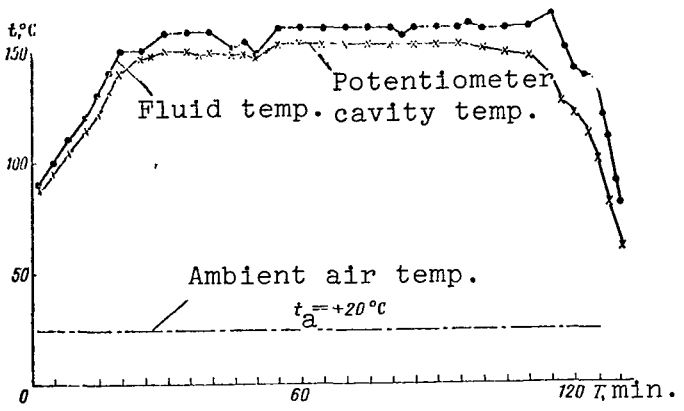

Figure 7.4. Experimental variation of servocontroller feedback potentiometer chamber temperature with thermal regime

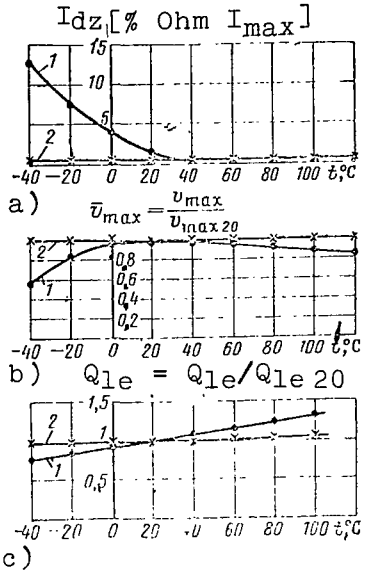

Figure 7.5. Dead zone (a), maximal velocity (b). and leakage flow (c) of servocontroller as functions of working fluid temperature (1) and ambient temperature (2)

However, in certain cases, the ambient temperature may have marked influence on hydraulic drive characteristics. For example, for boosters with complex kinematics, temperature deformation of the extended elements under the influence of the ambient temperature may lead to null shift, and the appearance of significant velocity asymmetry. With account for this discussion, we can hereafter examine only the influence of working fluid temperature on the hydraulic drive characteristics.

The working fluid parameter which varies most significantly with temperature is its viscosity (Figure 7.6). With temperature variation from + 20 to + 100° C or - 60° C, the viscosity of AMG-10 oil decreases by a factor of four, or increases by a factor of 165, respectively. The influence of temperature variation of the viscosity on output member maximal velocity is slight, since in this regime the

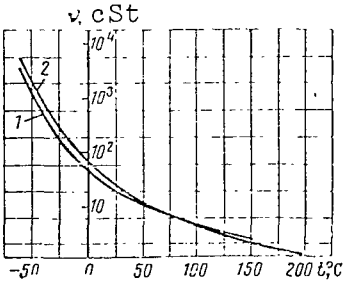

Figure 7.6. Working fluid viscosity versus temperature:

1 — for AMG-10; 2 — for 750S-3

working fluid flow in the hydraulic drive is usually turbulent, and the losses in the local resistances (primarily in the slide valve ports) change only slightly. However, viscosity change influences the internal passage resistance along their length, leading to reduction of the maximal output member velocity at − 30 - − 40° C by 5 - 15%, while at 100 - 150° C, it leads to increase by 2 - 5% in comparison with the values for +20° C. At − 40 - − 60° C, the degree of influence of viscosity increase on output member velocity increases markedly because of formation of the transitional and laminar fluid flow regimes in many of the passages and local resistances. The temperature variation of the viscosity may influence the leakage magnitude considerably, since the fluid outflow through the slide valve pair clearances is nearly laminar. /146

The working fluid density also changes noticeably as a function of temperature (Figure 7.7), causing reduction of the output member velocity at negative temperatures, and increase at high temperatures. In the case of hydraulic drive operation using AMG-10 fluid, with temperature variation from + 20 to + 100° C or to − 60° C, the output member velocity increases by 4% or decreases by 2.5%, respectively, because of the influence of fluid density change.

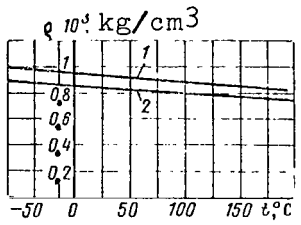

Figure 7.7 Working fluid density versus temperature:
1 — for AMG-10; 2 — for 750S-3

With temperature increase, both the working fluid compressibility (Figure 7.8) and air solubility in the working fluid increase. As a consequence of the air solubility increase, the amount of undissolved air which is evolved at points of reduced pressure may increase. Increase of the working fluid compressibility and amount of undissolved air lead to deterioration of hydraulic drive dynamic characteristics at high temperatures.

A second consequence of temperature variation is change of the geometric dimentions of the hydraulic drive structural elements. The slide valve pair elements are usually fabricated from the same material, therefore, the clearances in the slide valve pairs to not change with temperature change. Various metals are used in aircraft hydraulic drives: steel, aluminum alloys, titanium, bonnze, and so on. The coefficients of linear thermal expansion β for these metals are different (for steel, $\beta = 12 \cdot 10^{-6}$ 1/°C; for duraluminum, $22 \cdot 10^{-6}$ 1/°C, etc.). This difference may lead to change, with temperature variation, of the clearances between the hydraulic drive elements which were set at normal temperature (for example, the distance between the nozzle and flapper in the servocontroller hydraulic amplifier), or the neutral positions of the control elements (for example, the slide valve of the booster distributor unit). If the distributor housing is fabricated from aluminum alloy and the nozzle from steel, as a result of their different thermal deformations, the distance between the nozzle and flapper increases with temperature increase, and decreases with temperature reduction. Therefore, as a result of thermal deformations, with temperature increase the slope of the hydraulic amplifier pressure differential characteristic, distribution valve displacement, and output member velocity decrease, while all these characteristics increase with temperature reduction. The degree of this influence depends on the construction of the distribution device and nozzle, the magnitude of the clearance between the nozzle and flapper, and also the hydraulic amplifier setting selected. In the subject case, with temperature variation from + 20° to + 100° C, or to − 60° C, the output member velocity may decrease or increase, respectively, because of thermal deformations, by about 10%.

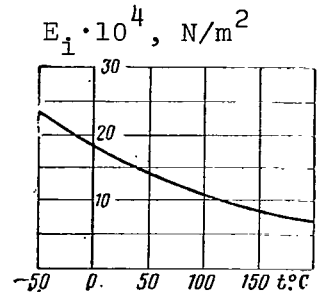

Figure 7.8. AMG-10 working fluid elastic modulus versus temperature for p = = 2 kN/cm^2

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Temperature change also has a marked influence on the power and mechanical characteristics of the servocontroller electromechanical transformer, leading to reduction of flapper displacement and reduction of output member velocity with temperature increase. The reverse pattern is observed with temperature reduction. With temperature increase from + 20 to + 100° C, the change of the electromechanical transformer characteristics causes a 5 - 10% reduction of the servocontroller output member velocity.

Thus, change of the working fluid viscosity and density, the /148
electromechanical transformer characteristic, and the thermal deformations of the hydraulic drive components have different effects on the output member velocity with temperature variation. The overall influence of these factors leads to reduction of the hydraulic drive output member velocity at negative temperatures. With temperature increase the output member velocity for boosters usually increases somewhat, while for servocontrollers this velocity may either decrease or increase, depending on the electromechanical transformer used and the structural characteristics of the servocontroller. Typical variations of booster and servocontroller output member maximal velocity with working fluid temperature are shown in Figures 7.9 and 7.10. As a consequence of the aforementioned temperature influence, the output member velocity should be checked at normal, maximal, and minimal working fluid temperatures.

High temperatures do not have any marked influence on hydraulic drive deadband, although for temperatures below 0° C the deadband may increase somewhat (Figure 7.5). Hydraulic drive leakage increases with temperature increase; therefore, maximal leakage should be checked only at the highest temperature. Temperature has practically no influence on the force developed. However, temperature
increase may lead to increase of the booster output member static /149
deflection because of increasing leakage in the distributor slide valve and reduction of its force characteristic slope.

One of the basic manifestations of the temperature influence is change of the properties of the rubber sealing rings and sleeves.

The various constructions of moving and stationary seals of aircraft hydraulic units have been examined in detail by Bashta, Razdolin, and others [3, 36]. In the present study, we examine only some characteristics of the most common types of moving rubber seals for piston rods, since they determine hydraulic drive hermeticity.

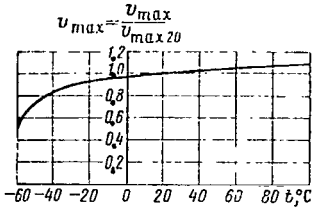

Figure 7.9. Maximal booster velocity versus temperature when using AMG-10 fluid

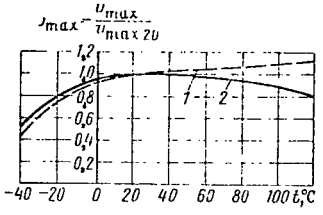

Figure 7.10. Typical servo-controller maximal velocity versus temperature when using AMG-10 fluid:

1 — with dominant influence of viscosity changes; 2 — with dominant influence of thermal deformations

Rubber O-rings are the most widely used seals. These rings are made from various rubbers having hardness up to Shore 90, and capable of withstanding pressure up to 3.5 kN/cm^2. They are used at temperatures from − 70 to + 200° C.

A schematic of O-ring sealing of the sliding joint of a piston rod and bushing with rectangular groove is shown in Figure 7.11a. The sealing ring is protected against extrusion into the clearance between the bushing and rod by a teflon backup ring.

The relative ring compression w is defined by the difference of the ring cross section diameter d in the free condition and the compressed ring nominal height h (in the radial direction) /150

$$w = \frac{d-h}{d} 100\%,$$

Usually, the quantity w for aircraft hydraulic drives amounts to from 10 to 20% [36]. Most rubbers swell by 5 - 10% in the working

fluids, which leads to additional increase of the ring compression and the necessity for increasing the groove size.

Sleeve-type seals, a schematic of one of which is shown in Figure 7.11b, are also used to seal piston rods. The advantage of these seals is minimal friction under no-load and low-load conditions. In the sleeve-type seal hermeticity is provided basically not as a result of precompression, but rather as a result of clamping of the sleeve "lip" contact surface by fluid pressure.

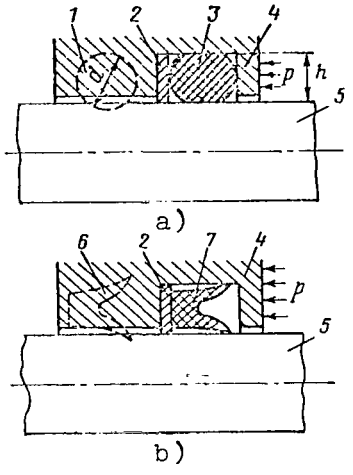

Figure 7.11. Schematic of piston rod seal:

a — O-ring seal; b — sleeve seal; 1 — O-ring in original condition; 2 — teflon back-up ring; 3 — O-ring in operating condition; 4 — bushing; 5 — piston rod; 6 — sleeve in original condition; 7 — sleeve in operating condition

Sealing rings and sleeves made from V-14, V-14-1, 98-1 and other rubbers, are used for operation in the temperature range from − 60 to + 100° C. For operation at higher temperatures, the IRP-1078, IRP-1054, 3465, and other rubbers (MRTU 38-5-1166-64) are used. However, the high-temperature seals do not ensure hermeticity at temperatures below − 40° C. Therefore, dual seals consisting of sequentially installed rings (or sleeve and ring) made from cold-resistant and heat-resistant rubbers are used for operation over a wide temperature range, from − 60 - + 100° C. Sometimes dual seals made from the same rubber are also used to ensure higher hydraulic drive reliability.

The elastic properties of rubbers deteriorate markedly with temperature reduction. Curves of O-ring (d = 9 mm, D = 60 mm) contact pressure as a function of temperature for typical heat-resistant and cold-resistant sealing rings with w = 16.5%, obtained by Nosov [30], are shown in Figure 7.12. The contact pressure

disappears completely for the 3465 rubber (curve 2) at a temperature of − 45° C. For the V-14 rubber (curve 1), the contact pressure remains constant down to − 15 - − 20° C, and then falls off rapidly, and at − 60° C amounts to about 10% of the initial value.

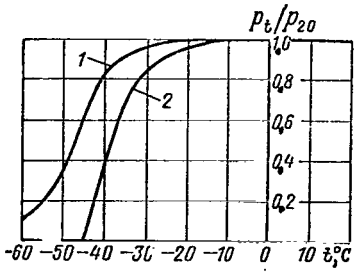

Figure 7.12. O-ring contact pressure versus temperature [30]

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The contact pressure variation depends on the degree of ring precompression [36]. With increase of the precompression, the absolute contact pressure reduction with temperature reduction increases, and the relative contact pressure reduction with temperature reduction decreases. It was also noted in [30] that, with gradual temperature reduction and long-term exposure, the contact pressure begins to decrease at a temperature 5 - 10° C higher than shown in Figure 7.12. When evaluating the influence of temperature on rubber O-rings and sleeves, we must consider that the coefficient of rubber linear expansion is about 10 times that of the metals, and amounts to $(8 - 15) \cdot 10^{-5}$. This also leads to reduction of the contact pressure with temperature reduction, and increase of the contact pressure and friction force in the seal with temperature increase.

The question of the necessary scope of hydraulic drive hermeticity is very important. To resolve this question, we must determine the hydraulic drive operating regimes which are most critical from the viewpoint of sealing loss.

According to Bashta's data, one of the most critical regimes for hydraulic unit sealing is with the airplane parked at a temperature of − 15 - − 25° C, and with no pressure in the hydraulic system. Under this condition, hydraulic unit leakage is sometimes observed under the influence of the fluid column pressure in the hydraulic system (usually no more than 2 N/cm^2), even for those seals which pass their sealing inspection tests satisfactorily at − 60° C.

This phenomenon is a result of the fact that the possibility of leakage depends on both the seal contact pressure and the working fluid viscosity, which are affected oppositely by temperature reduction. /152 The most critical combination of seal contact pressure and working fluid viscosity occurs in the temperature range indicated above (-15 to -25° C). At lower temperatures, the seal contact pressure decreases; however, the fluid viscosity increases to an even greater degree, which creates more favorable conditions for ensuring hermeticity. When the hydraulic system is operating, the working fluid pressure increases the seal contact pressure, thus ensuring hermiticity.

With the hydraulic system operating, the maximal and minimal temperature conditons are most critical for sealing. The seal contact pressure is minimal at the lowest temperature (usually - 60° C), because of loss by the rubber of its elastic properties and because of thermal deformation. As Muller showed in [34], the working fluid leakage flow through the hydraulic drive output member piston rod seal during its back-and-forth movement increases with contact pressure reduction and fluid viscosity increase. Therefore, the leakage flow is maximal with output member displacement at the minimal temperatures. The danger of high temperatures for hydraulic drive sealing is associated with the low working fluid viscosity and the considerable influence of the seal structural element thermal deformations. However, a special check of hydraulic drive sealing need not be made at the maximal temperature, and we can limit ourselves to checking the sealing when checking the hydraulic parameters under these conditions.

Temperature may have a marked influence on the electrical and mechanical properties of hydraulic drive electrical components (electromechanical transformers, electromagnets, potentiometric or inductive feedback sensors, and so on). Copper winding wire insulated by high-strength enamel (for example, wire of the PETV type with polyethylene enamel insulation) is widely used in hydraulic drive electrical components.

Temperature variation leads to change of the electrical resistance of the winding wire metal. The temperature dependence of metal resistivity is defined by the expression: /153

$$\rho_t = \rho_0[1 + \alpha(t - t_0)],$$

where ρ_0 and ρ_t are the metal resistivities at temperatures t_0 and t; α is the temperature coefficient of metal resistivity (for copper $\alpha \approx 0.004$ 1/°C).

The wire electrical insulation must ensure electrical strength, guaranteeing the absence of breakdown under any operating contitions, and high insulation resistance, preventing tne appearance of high leakage current wiich may disrupt the operating regime of the component. With temperature change, the electrical and mechanical properties of electrical insulation materials change, and this influences insulation electrical strength and resistance. The magnitude of the breakdown voltage U_{br}, characterizing the electrical strength strength of a dielectric, remains practically constant at temperatures up to 50 - 100° C, and then decreases exponentially [25] with further temperature increase:

$$U_{br} = kd\, e^{-\frac{a}{2}t},$$

where k is a constant for a given dielectric and given supply voltage frequency; a is the temperature coefficient of electrical conductivity; d is the thickness of the dielectric.

The different nature of the behavior of this curve is explained by the fact that at low temperatures there is a region of electrical breakdown, while at high temperatures there is a region of electrothermal breakdown.

The magnitude of the resistance of the solid dielectrics decreases with temperature increase in accordance with the relation:

$$\rho = \rho_0 e^{B/T},$$

where ρ is the resistivity of the dielectric; ρ_0 and B are the constants which are characteristic for the given dielectric; T is the absolute temperature.

Thus, high temperatures constitute the most critical temperature regime from the viewpoint of the possibility of insulation electrical breakdown or reduced resistance. /154

Temperature increase causes deterioration of electromagnet force characteristics, due basically to increase of the magnetic energy losses. Therefore, if control electromagnets are used in hydraulic drives, their operation must be checked at the maximal temperatures.

Abrupt ambient temperature variations may affect aircraft systems and components, however, abrupt temperature variations are not usually observed for hydraulic drives, which are massive systems filled with the working fluid and have high thermal inertia. When required, cyclic temperature changes which are close to the operational variations are simulated in hydraulic drive service life tests.

The following basic parameters characterizing water vapor content in the air are used to evaluate the influence of moisture present in the air on hydraulic drives and the methods used to test for moisture resistance:

— absolute humidity, i.e., the mass (in grams) of water vapor present per unit volume (usually one cubic meter) of air;

— dewpoint temperature, i.e., the temperature which moist air takes when it is cooled to complete saturation in relation to a flat water surface;

— relative humidity, i.e., the ratio of the actual water content to the maximal possible content for the given temperature, expressed in percent.

For constant relative humidity values, the absolute humidity increases sharply with temperature increase (Figure 7.13).

As the temperature changes, the water vapor present in the surrounding air may condense on the surfaces of the hydraulic drive and its components, and have a disrupting influence on the coating and metals, and also on the electrical insulation materials of the electrical components.

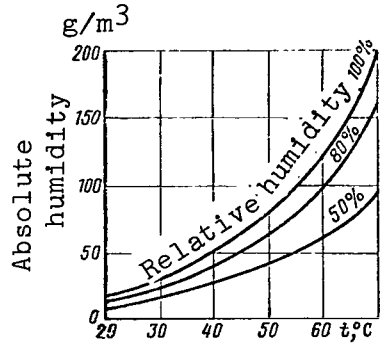

Figure 7.13. Absolute humidity of vapor-air mixture versus temperature for constant relative humidity

There are two types of processes of moisture interaction with materials: physicomechanical and chemical. In the physicomechanical interaction process, the moisture penetrates into the cracks, pores, and capillaries of the material, or is retained on irregularities. The electrical insulation materials and certain grades of plastics absorb moisture to the greatest degree. Moisture absorption and water film formation /155 on the surface of the electrical insulation material reduce its electrical strength and resistance, and may also lead to the occurrence of insulation arcover (arcover is considered to be a discharge in the liquid or gaseous dielectric adjacent to the solid insulation).

The chemical type of moisture interaction with materials leads to corrosion of the metals and breakdown of certain materials. Usually, we differentiate three types of corrosion: atmospheric, chemical, and electrochemical (galvanic). Electrochemical corrosion is due to the formation of galvanic pairs upon contact of different metals in an electrolyte medium. Chemical corrosion can occur if an aggressive nonelectroconductive medium contacts the surface of the metal. In the atmospheric corrosion case, the water vapor present in the atmosphere is adsorbed or condenses on the surface of the metal. A film of chemically pure water usually does not lead to any significant corrosion. However, water films can dissolve the

gases (particularly sulfur dioxide) and salt particles present in the atmosphere. The resulting electrolyte layer leads to more intense corrosion.

The corrosion rate depends on the relative humidity, temperature, thickness of the adsorbed or condensed water film on the surface of the metal, amount and composition of salt impurities in the water, and so on. The corrosion rate of metal increases with increase of the temperature and relative humidity, which is due basically to increase of the film thickness on the metal surface and the intensity of the electrochemical processes taking place. However, it should /156 be noted that the maximal corrosion rate occurs not with condensation of the largest possible amount of moisture, but rather for some worst amount of condensate (about 0.3 kg/m^2).

The action of corrosion degrades the commercial appearance of the hydraulic drive, and may lead to reduction of the mechanical strength of the metallic structural elements and to their premature failure. With high mositure content, there may also be degradation of the hydraulic drive paint coatings, and also of the bonded joints, leading to peeling of gaskets bonded to covers, debonding of nameplates, and so on. Intense electrochemical corrosion of the hydraulic drive surfaces may occur under the action of ocean fog containing a large amount of salts.

With increase of the air temperature and humidity, the dielectric properties of air deteriorate, which may lead to insulation breakdown or arcover in the hydraulic drive electrical components (for example, in microswitches, wiring panels, and electrical disconnects).

The action of moisture and ocean fog represents considerably less danger for hydraulic drives than for electrical and electronic equipment. These influences usually do not cause hydraulic drive failure, but rather only degrade their external appearance. The only real hazard for hydraulic drive serviceability will be

represented by deterioration of the electrical insulation electrical properties and the influence of corrosion on those hydraulic drive members operating under severe conditions which are not adequately protected against the action of corrosion.

Low atmospheric pressure has practically no effect on the hydromechanical characteristics of hydraulic drives, which are sealed and operate at high working pressures. However, reduced atmospheric pressure may influence the electrical strength of the insulation of hydraulic drive electrical components, in which the insulation strength is determined by the dielectric properties of the air. In the operating air pressure range, the dielectric strength of the insulating air gaps for a uniform electrical field is proportional to the air density and atmospheric pressure. The relative reduction of air gap electrical strength with increase of height H is shown in Figure 7.14. Considering the nature of the influence of air temperature and pressure on insulation electrical strength, we can conclude that under reduced atmospheric pressure conditions, the hydraulic drive electrical component insulation strength should be checked at the highest specification temperature.

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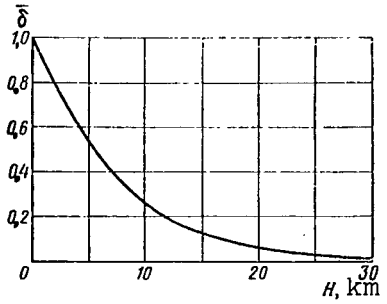

Figure 7.14. Relative air gap electrical strength versus altitude (for constant temperature)

The frost and dew which from on hydraulic drive surfaces have practically no effect on the units. Dew contributes to the formation of corrosion; however, the degree of its influence is considerably less than in the case of long-term action of high humidity during tests for moisture resistance.

Sand and dust may lead to the appearance of very fine scratches on the hydraulic drive output member piston rod surface, which leads to premature seal wear and leakage. Therefore, the hydraulic drives

should be mounted aboard the airplane in areas which are isolated to the maximum possible degree from the action of sand and dust.

Fungi which form on the hydraulic drive surfaces may cause corrosion of metals, breakdown of electric insulation materials, destruction of coatings, and so on. Insects (termites, red ants) or rodents (rats, squirrels, field mice, and so on) may present considerable hazards for the electrical cables and wiring. Materials and coating recommended by the specialized institutes (GOST 15963-70) must be used to counteract the influence of biological factors on hydraulic drives.

Solar radiation has no effect on hydraulic drives, since the /159
drives are protected against direct action of sunshine aboard the airplane. In most cases, rain also has no effect on aircraft hydraulic drives. Only in certain cases, when the hydraulic units are inadequately protected against contact with rain or spray during airplane taxiing and ground roll (for example, units on the nose gear strut), may it become necessary to conduct hydraulic drive tests for spray proofing. Spray can penetrate into electrical connectors or under sealing covers, leading to corrosion and failure of the hydraulic drive electrical components.

The recommended inspection tests of hydraulic drives for stability and resistance to the action of climatic and mechanical factors are presented in Table 7.1.

We note that the influence of high and low temperatures leads to significant irreversible changes of the hydraulic drive structural element properties. However, these effects are usually reproduced during service life tests.

TABLE 7.1.

Test made	Servocontrol	Booster
Vibration stability	P	—, H
Vibration strength	+	+
Shock strength	+	+
Shock stability	—, I	—, I
Resistance to linear accelerations	P	—, I
Shipping strength	+, H	+, H
	only in series production	
Resistance to acoustic noise	—, I	—, I
Thermal stability	+	+
Cold resistance	+	+
Resistance to cyclic temperature change	—	—
	considered during service life tests	
Resistance to action of frost and dew	—, I	—, I
Moisture resistance	+	+
Resistance to reduced atmospheric pressure	P, H	—, I (P, H when electric components present)
Resistance to ocean fog	— (P, H when required)	—
Dust proofing	—, I	—, I
Spray proofing	— (P, H when required)	—, I
Resistance to solar radiation	—, M	—, M
Resistance to the action of biological factors	—, M	—, M

Note. "+" — tests are conducted; "—" — tests are not conducted; "P" — one-time tests are conducted; "H" — tests are conducted with unit inoperative; "I" — confirmed by operation on article; "M" — confirmed by material selection.

7.2. Equipment for Climatic Tests

Tests of hydraulic drives for thermal stability at working fluid and ambient temperatures up to + 100° C are conducted on conventional test stands. The required working fluid temperature increase is achieved by throttling the working fluid flow through the bypass valve or unloading valve with the radiator out of the circuit. In this case, all the components of the test stand hydraulic system must permit operation at high temperatures. When it is necessary to simulate high ambient temperature, we can use hot boxes and air blowers with electrical heating installed directly on the test stand frame.

Special thermal stands are used to simulate temperatures above + 100° C. The most complex system in the thermal stand is the hydraulic supply system; the system for simulating high ambient temperature is considerably simpler, and the load simulation system is the same as that used on the conventional stand. /160

Electrical heating is most frequently used to obtain high working fluid temperature. A special heating section, whose ends are separated from the line by insulators, is installed in the hydraulic supply system line. The working fluid in the hydraulic system is heated as it flows through the heater. A schematic of the hydraulic system of a very simple thermal stand is shown in Figure 7.15. Installation of the heater in the delivery line makes it possible to reduce the maximal working fluid temperature in the pump required to create the required temperature at the entrance to the hydraulic drive being tested. In this case, the working fluid temperature in the pump is below the required temperature at the hydraulic drive inlet. When the heater is installed in the return line, it is easier to ensure sealing of the electrically insulated heater segments, and it is possible to use thinner-wall tubing in the heater; however, in this case, the pump must operation with working fluid temperature exceeding that required, and this may lead to pump failure. Therefore, it is still advisable to install the heater in the delivery line in such thermal stands as well. The temperature

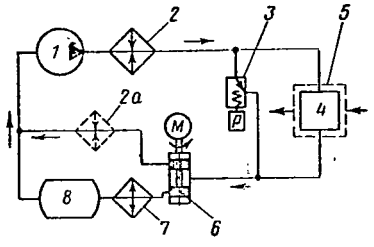

Figure 7.15 Schematic of thermal stand hydraulic supply system:

1 — pump; 2 — electrical heater in supply line; 2a — electrical heater in return line; 3 — bypass valve; 4 — test hydraulic drive; 5 — thermal chamber; 6 — selector valve; 7 — oil-water radiator; 8 — tank

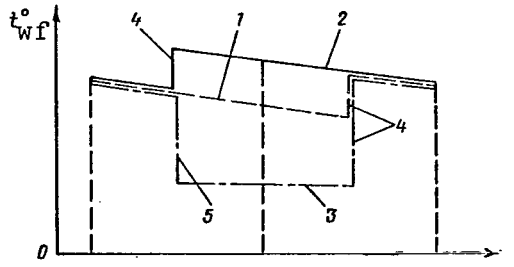

Figure 7.16. Temperature variation along hydraulic supply system circuit:

1 — with heater installed in supply line; 2 — with heater installed in return line; 3 — in scheme with "cold" pump; 4 — temperature rise in heater; 5 — temperature drop in radiator

variation along the hydraulic system circuit in the steady-state thermal regime for different heater installation locations is shown in Figure 7.16 (curves 1 and 2).

In such thermal stand schemes, considerable difficulty may be encountered in ensuring long-term serviceability of the pump with the high working fluid temperature. Thermal stands with "cold" pump are used to overcome this difficulty. In order to obtain low /161 working fluid temperature in the pump, a powerful heater can be installed in the supply line, and a powerful cooler in the return line. The fluid temperature variation along the circuit of such a thermal stand is shown in curve 3. A drawback of this scheme is the large electrical power required for the heater, and the necessity for using a large radiator, which requires high water consumption.

A hydraulic system with pulsating flow [46] can also be used as the hydraulic supply source for the thermal stand with "cold" pump. In this hydraulic system, a hydraulic divider and hydraulic

rectifier are used to provide different temperatures in the hydraulic system hot loop and in the pump unit (Figure 7.17). The absence of fluid recirculation in the hydraulic system makes it possible for the pump to operate at moderate temperature even if the working fluid temperature in the hot loop is very high. The back-and-forth movement of the fluid mass about the middle position in the pump unit delivery lines is transformed with the aid of the rectifier check valves into unidirectional flows in the hot loop supply and return lines. Hydraulic accumulators are installed in the hot loop supply and return lines to reduce the pressure pulsations. Pumps of constant or variable output, operating with constant delivery pressure and an automatic switching valve which creates the pulsating flow in the pump unit discharge lines, can be used in the pump unit. A drawback of this scheme is the difficulty in providing the high required flow rates, and the presence of the pressure pulsations.

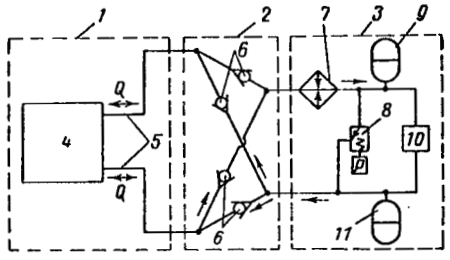

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Figure 7.17. Schematic of thermal stand with pulsating flow:

1 — cold pump loop; 2 — hydraulic rectifier; 3 — hot loop; 4 — pump unit with divider, which creates pulsating flow; 5 — line; 6 — check valves; 7 — heater; 8 — bypass valve; 9 — hydraulic accumulator in supply line; 10 — test hydraulic drive; 11 — hydraulic accumulator in return line

The thermal stand electric heaters usually operate in the relay mode; in this case, the current is turned on and off automatically or manually so as to ensure the required temperature variation, or so as to maintain the required temperature. The heater power must provide the required maximal temperature rise rate.

When designing the heater and selecting its operating conditions, we must be certain that the maximal heater tube wall temperature does not exceed the maximal admissible working fluid temperature

at which intense breakdown of the working fluid begins. It is necessary to provide an automatic interlock to prevent the possibility of activating the electric heater without working fluid flow through the heater. The heater is usually a spiral tube 50 - 100 m long, installed in a thermally insulated container. /163

If it is necessary to simulate the ambient temperature, the test hydraulic drive is located in the stand thermal cabinet and uniformly heated by the air coming from the blower, after being heated to the required temperature. Air temperature variation is accomplished either by regulating the current in the electric heater spiral or by means of a vane which regulates the ratio of two streams with fixed high and normal temperatures in the air stream traveling to the hydraulic drive.

The output member of the hydraulic drive mounted in the stand thermal chamber is connected by a link which passes through the wall of this chamber with a conventional loading device. In order to minimize heat losses, the hydraulic lines in the stand should be as short as possible and thermally insulated.

Testing of hydraulic drives at negative temperatures requires a stationary cold stand, which makes it possible to install the system inside the test stand and permits test personnel to work inside. Considerable difficulties are encountered in the development of a hydraulic system which permits maintaining the required negative working fluid temperature for a long time period. It is advisable to locate the entire hydraulic supply system, including the tank and pump, inside the cold chamber. A schematic of a hydraulic supply system for cold stability tests is shown in Figure 7.18. The radiator 8 is a spiral tube with large cooling surface area, which can be cooled by cold air from the blower. Rapid cooling of the working fluid in the hydraulic system can be accomplished by opening the selector valve.

It is convenient to drive the constant-output hydraulic pump from a hydraulic motor fed from an auxiliary external hydraulic

system. The minimal output of the stand hydraulic system at which the required working fluid pressure and flow rate are developed can be established by regulating the hydraulic motor rotational speed. It is more convenient to locate the hydraulic motor inside the chamber and pipe the hydraulic supply to it from the auxiliary hydraulic system through fittings in the cold chamber wall. The pump can also be driven by an electric motor located outside the chamber through a mechanical adapter in the chamber wall.

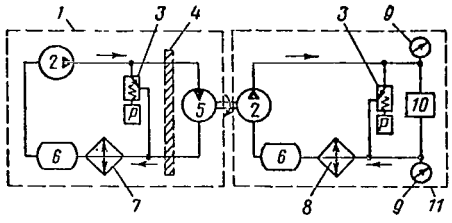

Figure 7.18. Schematic of hydraulic supply system for cold stability testing:

1 — hot loop; 2 — pump; 3 — bypass valve; 4 — cold chamber wall; 5 — hydraulic motor; 6 — tank; 7 — oil-water radiator; 8 — oil-air radiator; 9 — pressure gauge; 10 — test hydraulic drive; 11 — cold loop

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Since the line resistance increases considerably at negative temperatures, it is necessary to install monitoring pressure gauges in the supply and return lines, no further than 20 cm from the hydraulic drive. We note that accuracy class 0.5 pressure gauges are not suitable for measuring pressure at ambient temperatures from − 60 to − 50° C, and gauges of accuracy class no higher than 1.5 must be used.

Temperature measurement accuracy is very important when testing for heat stability and cold stability. In the case of variable thermal regimes, it is necessary to consider thermocouple thermal inertia, since some time is required for heating (or cooling) the thermocouple bulb and the thermosensitive junction. Thermocouple inertia can be characterized by the thermal time constant, which for the thermocouples usually used amounts to 5 - 15 seconds. The thermal time constant for thermocouples which measure air temperature is 2 - 3 minutes. Thermocouple inertia must be considered when /165
checking hydraulic drive parameters in a narrow range of specified temperatures with comparatively rapid change of the actual temperature. This case is encountered frequently in tests for cold stability of hydraulic drives with high required flow rate.

Other types of climatic tests are conducted in the universal chambers described in [8, 15, 25].

7.3. Hydraulic Drive Climatic Test Technique

The techniques for determining the basic parameters of boosters and servocontrollers during heat stability and cold stability tests are basically the same as under normal conditions. When, in testing hydraulic drives for heat stability, is is possible to limit ourselves to creating the required working fluid temperature (with normal ambient air temperature), it is necessary to create in the hydraulic system the required working fluid temperature and then, in the course of 8 - 10 minutes, create the required hydraulic drive thermal regime by stroking the output member. When it is necessary to reproduce the ambient air temperature, we must maintain the hydraulic drive at the specified ambient temperature for no less than two hours prior to feeding the hot working fluid into the hydraulic drive.

In the case of hydraulic drive cold stability tests, we must mount the hydraulic drive on the stand in the cold chamber, and maintain it at the specified ambient temperature for no less than four hours (no less than six hours for hydraulic drives weighing more than 10 kg). With working fluid temperature – 60° C, in many cases it is necessary to check for trouble-free startup, functioning, and sealing. For this check, the stand hydraulic system is activated, and the required pressure is built up at the hydraulic drive inlet. Then, in the course of 8 - 10 minutes, the hydraulic drive output member is operated through its travel with dwell times of 5 - 10 seconds in each extreme position. The leakage magnitude should not exceed the specified magnitude (or no leakage). Verification of hydraulic drive parameters during cold stability tests /166 are usually made at temperatures above -- 60° C (for example, at – 20° C.

Verification of hydraulic drive sealing should also be made at ambient temperature – 15° C with pressure 2 N/cm^2 applied to all the hydraulic drive fittings, developed by a hand pump. The hydraulic drive soak time in these conditions should be at least four hours.

When conducting heat stability and cold stability tests, the working medium and ambient temperatures should be maintained to within $\pm$ 3° C. The hydraulic drive is considered to pass the heat stability and cold stability tests if, in the process of these tests and after the tests (under normal conditions), the hydraulic drive parameters correspond to the specification values.

It is convenient to combine booster frost and dew tests with the cold stability tests. After finishing the cold stability tests, the booster should be taken from the cold chamber, quickly installed on the test stand, and the breakaway force and dead band checked every 20 - 30 minutes over a period of three hours.

The techniques for conducting the other climatic tests are the same as for other articles [15, 25]. During these tests, hydraulic fluid is not supplied to the hydraulic drives. During tests of servocontrollers for moisture stability and altitude performance, a moisture resistant cable leading outside the chamber is connected to the electrical connectors.

During altitude tests of servocontrollers, after maintaining the unit at the required reduced pressure (for example, at 0.2 N/cm^2) and temperature + 60° C, the electrical circuits are checked for breakdown.

Let us examine in more detail the technique for testing hydraulic drives for moisture stability, and the questions of accelerating these tests. Moisture stability tests are conducted in one of two ways: either by continuous maintenance of the specified temperature and relative humidity, or by cycles with daily forced dew precipitation.

In the tests using the first technique, after inatalling the hydraulic drive in the moisture chamber, the temperature in the chamber is raised to the specified value (for example, to 40 ± 0.5° C) and, after 1.5 - 2 hours with the unit at this temperature, the relative humidity is increased to 95 ± 3%. This approach to the specified regime prevents extensive precipitation of the dew which is formed with simultaneous increase of the temperature and humidity. Then the specified regime is maintained constant for the specified number of days. During these tests, droplet formation and droplet deposition on the surface of the unit are not permitted, only slight deposition of dew in the form of sweat is permitted. Check of the resistance and strength of the hydraulic drive electrical circuit insulation should be made either in the moisture chamber using a moisture resistant electrical cable, or outside the chamber no longer than three minutes after taking the hydraulic drive from the chamber. If the duration of the checks is more than three minutes, the unit is placed back into the chamber and maintained there for two hours, after which the checks can be continued. The external appearance of the hydraulic drive is checked after removal from the moisture chamber. /167

Tests for moisture stability using the second technique are conducted in 24-hour cycles. During each cycle, forced dew deposition is ensured by one of the following methods: a) turning off the chamber heat and moisture sources for 6 - 8 hours; b) reducing the air temperature by 5 - 10° C for no more than one hour, with subsequent rapid temperature increase; c) by supplying warmer (by 5 - 10° C) moist air to the chamber [25]. The first of these moisture stability testing techniques is the simplest and most widely used.

In accordance with GOST 15151-69, after the moisture resistance tests, individual distortions of the paint coating are permitted, which disappear after 12 - 24 hours under normal climatic conditions, or individual corrosion sites are permitted if this does not affect the operability and reliability of the units, and does not degrade their external (commercial) appearance.

Since the moisture stability tests are lengthy, the question of their acceleration is important. In developing an accelerated technique for these tests, we must be certain to maintain the nature of the moisture destructive action process.

The simplest way to accelerate the tests lies in raising the /168
temperature while keeping the same relative humidity, and also reducing the minimal cycle temperature (for the cyclic regimes). In this case, the absolute humidity increases, the rate of the corrosive electrode reactions increases, and the thickness of the condensate film on the surface of the item being tested increases when reducing the temperature in the course of each cycle. The equivalent accelerated regimes for tests in the continuous and cyclic regimes are presented in GOST 15963-70.

Several other accelerated moisture resistance testing techniques are also in use. The corrosion rate can be increased considerably if the condensed moisture film is periodically dried and condensation is again permitted, if we increase the salt content in the water vapors, and so on. However, it is very difficult to find the exact scaling coefficients for all these techniques; therefore, they can be recommended only for conducting comparative moisture resistance tests of articles.

CHAPTER VIII

HYDRAULIC DRIVE SERVICE LIFE TESTS*

8.1. Influence of External Factors on Hydraulic Drive Service Life

In the process of hydraulic drive operation, various irreversible changes of their structural components (or design-technological factors) take place as the drives accumulate service time. The primary destructive factors leading to the irreversible change accumulation are wear, aging, and fatigue damage accumulation. The processes of irreversible change accumulation in hydraulic drive structural components can lead to the appearance of either catastrophic (sudden) or parametric (gradual) failures. An example of sudden failure is fatigue failure of the hydraulic drive output member or actuator cylinder housing as a result of fatigue damage accumulation in these parts. A parametric failure is, for example, the onset of excessive external leakage because of permanent deformation accumulation and contact pressure reduction in rubber seals. /169

Figure 8.1a shows the typical process of irreversible change accumulation L (t) in the weakest component of a hydraulic drive. This process (as well as the processes of irreversible change

*Chapter written together with Yu. V. Tochilin.

accumulation in the other components) is random, and depends on a combination of external factors during operation or test stand service life tests, and also on the design and technological errors in fabrication of the unit. The magnitudes of the irreversible changes L_1^* and L_2^* characterize the range of L values within which failures due to the action of the subject irreversible change may occur. The failure probability density function f_1 corresponding to this change has the form shown in Figure 8.1b. The processes of irreversible change accumulation in the other structural components of the hydraulic drive may lead to the occurrence of failures for which the distribution density functions (f_2, f_3, and so on) may either cross the function f_1 (characterizing the weakest component) or may not cross the latter. The maximal achievable guaranteed hydraulic drive service life is defined by the function f_1 for the weakest component of the hydraulic drive. During hydraulic drive overhaul after running out the guaranteed service life, the service life to first overhaul, or the service life between overhauls, one or more of the weakest hydraulic drive components are reworked or replaced. In this case, the hydraulic drive technical service life is determined by the process of irreversible change accumulation and by the failure distribution density function of the weakest nonrepairable hydraulic drive component or by economic considerations.

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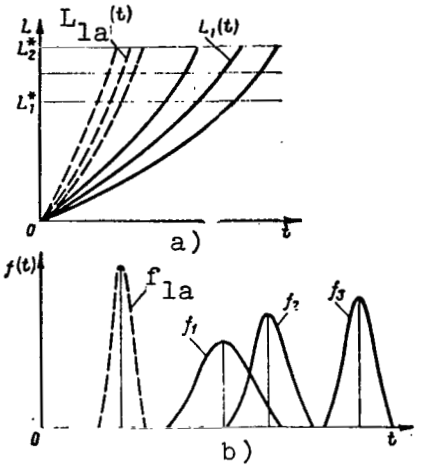

Figure 8.1. Irreversible change accumulation (a) and failure distribution density (b) as functions of operating time in the operational regime and in accelerated tests

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Let us examine the physics of the irreversible change accumulation processes in hydraulic drive structural components. The primary and most complex form of destructive imput is wear, which leads to the accumulation of permanent changes in the moving joint seals, slide

valve pairs, hinged joints of booster kinematic members, and the hinged joint of the output member with the load.

By hydraulic drive component wear, we mean the process of destruction of the surface layers of parts during friction or during inputs which lead to onset in the parts of mechanical, thermophysical, or chemical processes. Wear of parts leads to change of the dimensions and shape of the parts and change of their surface properties.

On the friction surfaces of parts and at point of their contact there may develop fatigue phenomena, elastic and plastic deformations of the microroughnesses, mutual molecular pickup of metal particles, heating of the parts to high temperatures, phase transformations, oxidation, formation of carbides, sulfides, and other chemical compounds.

However, one of the processes noted above usually is dominant for each of the hydraulic drive friction pairs, and this process determines the part surface destruction mode and its wear rate, and is determined by the materials, shape and dimensions of the friction pair, magnitude and nature of the load, relative displacement velocities, temperature conditions, lubrication amount and quality, and so on.

It has been shown by Khrushchov [43] and Kragel'skiy [20] that the primary forms of wear are mechanical, molecular-mechanical (pick-up), and corrosional-mechanical wear of the friction surfaces of parts. Mechanical wear is characterized by surface destruction as a consequence of elastic and plastic deformations of the surface layers under the action of external mechanical influences without significant physical or chemical changes.

The basic subforms of mechanical wear are fatigue and abrasive wear. In the fatigue wear case, the frictional bond (i.e., bond of the microroughnesses of the friction surfaces) is destroyed as a /172

consequence of either elastic or plastic displacement and, in the abrasive wear case, as a consequence of microcutting.

In the fatigue wear case, surface destruction takes place as a consequence of repeated mechanical actions on a given microsegment of the friction surface.

Wear can be characterized by the specific wear and by the number of cycles to destruction of the friction bond. The specific wear is the amount of material worn along a path equal to the diameter of a single contact spot divided by the actual contact area and by the average diameter of a single contact spot.

The specific wear and number of cycles to destruction of the friction bond differ markedly, depending on the wear mode.

The specific wear is minimal for the elastic displacement case, amounting to $10^{-5} - 10^{-8}$; for plastic displacement it is $10^{-3} - 10^{-4}$; and for microcutting — $10^{-1} - 10^{-2}$. The number of cycles to failure in the elastic displacement case is $10^{4} - 10^{7}$, for plastic displacement — $10^{2} - 10^{3}$, and for microcutting — 1.

In hydraulic drives, the components most subject to wear are the output member (piston rod) rubber seals, which operate under worse conditions than the piston seals or the other seals, which are exposed to the working fluid on both sides. The nature and magnitude of the wear depend on the elastic and strength properties of the wear material, seal construction, condition of the working surfaces of the seal components, prestretch of the sealing ring, working fluid pressure, temperature, piston rod velocity, and working fluid type. Depending on the combination of the aforementioned factors, there may be boundary (dry), mixed, or fluid (hydrodynamic) friction in the seal.

Boundary friction can occur in seals at very low piston rod velocities (from 0 to 10 mm/sec), in which case the friction obeys the Coulomb law and the friction coefficient is independent of the velocity [47]. The mixed friction regime develops at higher piston rod velocity. In this regime, the friction coefficient decreases /173
monotonically with piston rod velocity increase. In general form, we can say that, with decrease of the operating pressure and increase of the piston rod velocity, the boundary friction fraction diminishes and the fluid friction fraction increases.

The fluid friction regime begins when the load on the seal is taken completely by the working fluid layer and there is no contact between the seal ring and piston rod surfaces. In the presence of fluid friction, hydrodynamic pressure develops between the working surfaces of the rod and ring as the rod moves relative to the sealing ring. When the hydrodynamic pressure reaches the magnitude of the compressive radial pressure between the rod and the sealing ring, a radial annular gap develops through which the working fluid flows (seepage occurs).

The curve of static pressure distribution along the sealing ring contact surface in the presence of operating pressure has an asymmetric form; therefore, the pressure gradient at the point of entry into the radial gap and, consequently, the lubrication conditions in the gap are significantly different for the forward and reverse piston rod strokes. The conditions of seal loading by the working pressure and friction on the sliding surface are also different during the forward and reverse strokes. During forward travel, the load is "compressing", while during reverse travel, it is "stretching". In the "stretching" load case, a larger pressure difference must be overcome than in the "compressing" load case. During the piston rod retraction stroke, there is available for lubricating the seal only a thin layer of oil on the rod, which was extended prior to this during the forward stroke. With reverse stroke velocity on the same order of magnitude as the forward stroke velocity, this amount of oil is not sufficient to create the

equivalent lubrication conditions in the clearance. As a consequence of this, the friction coefficient may have a higher value during the reverse stroke [47].

In the fluid friction case with back-and-forth-movement of the piston rod, the amount of fluid carried out in the forward rod stroke exceeds the amount of fluid brought back into the unit in the reverse stroke. In [34], Muller obtained an expression for the amount of fluid flowing out of the hydraulic drive along the piston rod in a single double stroke: /174

$$Q_{le} = Q_{fwd} - Q_{rev} = \frac{\pi\sqrt{\mu}}{2} \times$$

$$\times DH \left(\alpha_{fwd} \sqrt{\frac{v_{fwd} l_1}{p_i^*}} - \alpha_{rev} \sqrt{\frac{V_{rev} l_2}{p_{rev} + p_i^*}} \right), \qquad (8.1)$$

where H is the rod travel; α_{fwd}, α_{rev} are the coefficients of film thickness for the forward and reverse strokes; v_{fwd}, v_{rev} are the rod velocities in the forward and reverse strokes; l_1, l_2 are the widths of seal contact with the piston rod in the forward and reverse strokes; p_i^* is the maximal sealing ring pressure on the piston rod (minus the working fluid pressure); p_{rev} is the working fluid pressure during the reverse stroke.

From examination of (8.1), we find that the leakage is directly proportional to the piston rod diameter D and its stroke. Leakage increases with increase of the fluid pressure p_{rev}, since in this case the amount of fluid returning to the hydraulic drive chamber in the reverse stroke decreases. The leakage increases as the square root of the viscosity μ and the hydraulic drive piston rod velocity. When using a harder material or greater initial deformation, the maximum of the pressure p_i^* will increase, which leads to less leakage.

We note that (8.1) is valid if the smallest hydrodynamic film thickness is greater than the height of the sealing ring and piston rod surface microroughnesses. However, this case is encountered very rarely in practice.

Thus, in the case of fluid friction in the seal, for which seal wear is minimal, there must be leakage along the rod. Therefore, slight leakage along the output member is necessary in order to ensure long hydraulic drive service life. If complete hydraulic drive sealing is required, the hydrodynamic seal regime is not allowable, and we must use seals operating basically in the mixed friction regime, which leads to more rapid seal wear and considerable service life reduction.

The piston seals which are wetted by the working fluid on both sides operate under considerably better conditions. For the piston sealing rings operating with a "stretching" load, the friction decreases markedly with increase of the piston velocity, which is explained by the good lubrication conditions. In the "compressing" load condition, the hydrodynamic regime arises at a low velocity [47]. Therefore, piston seal wear is usually considerably less than piston rod seal wear. /175

In order to determine the dependence of piston rod seal friction force on the hydraulic drive operating regime, we conducted a study for the most common seal types on a setup in which a 30-mm diameter piston rod was displaced in a hydraulic cylinder by a servocontroller connected with the piston rod through a strain-gauge force sensor. The required pressure in the hydraulic cylinder chamber was established by a hand zero-flow hydraulic press. The required working fluid temperature in the hydraulic cylinder was obtained by maintaining the fluid for three hours at the required ambient temperature created by means of an air blower and electric heater or dry ice. The hydraulic cylinder piston rod and the force sensor were articulated by means of a hinged joint in order to exclude any possible misalignment of the piston rod in the sleeves. The rod

displacement (for determining velocity), friction force, and fluid pressure in the hydraulic cylinder were recorded on a light-beam oscillograph. In view of the symmetric positioning of the two test seals, the measured friction force is the average of the forward and reverse travel friction forces. The tests were conducted for piston rod velocities from 5 to 200 mm/sec, pressures from 0 to 2000 N/cm^2, temperatures from − 45 to + 100° C, and using AMG-10 working fluid.

The curves of friction force versus velocity obtained under the different conditions for circular rings (with w = 16.5% precompression) are shown in Figure 8.2. We see from these curves that the

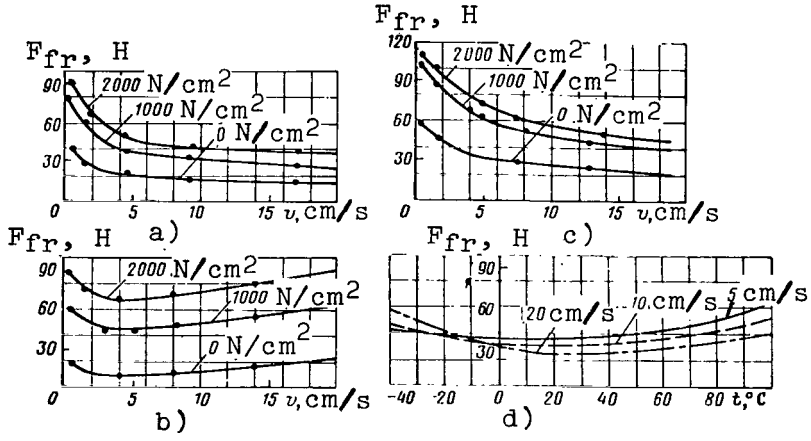

Figure 8.2. Piston rod seal friction force as function of operating regime:

a — friction force versus velocity and pressure for + 20° C; b — friction force versus velocity and pressure for − 45° C; c — friction force versus velocity and pressure for + 100° C; d — friction force versus temperature for pressure 1 kN/cm^2

seal friction is basically of the mixed type. Only for negative working fluid temperatures do we observe transition from mixed friction to fluid friction at a velocity of 4 - 5 cm/sec. This is due to working fluid viscosity increase and sealing ring contact pressure decrease with temperature reduction. With temperature increase, /176

the friction force increases, which is caused by the large contact pressure increase because of sealing ring thermal expansion. The friction force increase at negative temperatures is due to initiation of the fluid friction regime. With increase of the working pressure, the friction force increases nearly linearly.

In the mixed friction zone, the dependence of seal friction force on piston rod velocity may be described approximately by the expression:

$$F_{fr} = k_f v^{-b}, \tag{8.2}$$

where k_f is a coefficient which depends on the seal type and size, rubber grade, and seal operating regime; b = 0.1 in the velocity range from 0.5 to 20 cm/sec for sinusoidal output member velocity variation.

The results of [7, 21], and also analysis of seal wear in hydraulic drives showed that in the working range of operational temperatures, working pressures, and piston rod velocities, the seal wear of moving joints with elastic contact has a fatigue nature. /177

The wear of piston rod rubber seals can be evaluated on the basis of the amount of eroded material ΔV [20]:

$$\Delta V = iLS_a \tag{8.3}$$

where L is the friction path; S_a is the actual contact area; i is the specific wear.

Since the seal wear in hydraulic drives has a fatigue nature, the specific wear depends most strongly on the elastic and strength properties and surface condition of the wear material, and also on the friction coefficient f [20, 39]:

$$i = k_i f^a, \tag{8.4}$$

where k_1 is a constant which depends on the elastic and strength properties of the material and the surface condition; a is the rubber seal fatigue coefficient, constant for a given rubber grade.

Rybalov found in [39] that a = 2.4 for rubber based on the combined rubbers SKN-18 and SKN-26; for rubbers based on the SKF-32 rubber, a = 3.85; and for rubbers based on the SKEP rubber, a = 4.9.

Using (8.2) and (8.4), we can obtain the dependence of the specific wear i on the piston rod velocity v (for mixed friction in the seal)

$$i = k_{i1} v^{-ab}. \tag{8.5}$$

Normally, in the working fluid temperature range from + 20 to + 60° C, temperature variation does not have any marked influence on the magnitude of the specific wear. With temperature increase above + 100° C, the specific wear i increases, and this increase can be described by the equation [39]:

$$i_t = i_{100}\left(\frac{t}{100}\right)^{a_1}, \tag{8.6}$$

where a_1 = a.

In the temperature range from + 60 to + 100° C, we can consider /178 that the specific wear increases with temperature increase, following the equation:

$$i_t = i_{<60}\left(\frac{t}{60}\right)^{a_1}, \tag{8.7}$$

where $a_1 = a\,\frac{t - 60}{40}$.

The seal specific wear may be influenced markedly by the contact temperature Δt_k, by which we mean the local temperature rise in the contact zone owing to friction. Komarov showed in [19] that the contact temperature depends linearly on piston rod velocity, and may be described by the expression:

$$\Delta t_k = k_t v, \tag{8.8}$$

Where Δt_k is expressed in °C; v is the piston rod velocity in cm/sec; $k_t \approx 0.2$ (for rubber V=14 when w = 13% and p = 0).

The quantity k_t increases with pressure increase in the hydraulic drive power cylinder chambers. For the aircraft hydraulic drive piston rod seal types and parameters usually used, we can take approximately k_t = 1 °C · sec/cm.

The wear of hydraulic drive slide-valve components during their service life is usually very slight, and does not have any marked influence on the variation of the hydraulic drive output characteristics and serviceability.

In certain cases as operating time increases, there is some erosion of the silde valve pair working edges, owing to local cavitation and erosion in the maximal working fluid flow velocity zone. Erosion of the slide-valve edges may lead to some increase of the leakage, reduction of the deadband, reduction of the slope of the slide-valve pair pressure drop characteristic, and increase of the static deflection under load (for the booster); however, the degree of this influence is usually small.

A second destructive factor is thermal aging of the hydraulic drive structural components, due basically to the physical and chemical processes of oxidation in the materials. The aging processes of /179
all materials obey the Arrhenius law:

$$u = A_u e^{-\frac{E}{RT}}, \tag{8.9}$$

where u is the rate of the oxidation processes in the material; A_u is a constant; E is the activation energy of the aging process; T is absolute temperature; R is the gas constant.

Aging has the greatest influence on deterioration of the properties of the deformed rubber sealing rings and sleeves, reducing the contact pressure and increasing the permanent deformations, which may lead to hydraulic drive sealing loss.

The use of (8.9) to calculate the seal aging processes and determine seal life (in regard to aging) is very difficult, since the magnitude of the activation energy is usually not known exactly. Therefore, in practice, we usually use, for the calculation, the experimental curves of fatigue deformation accumulation as a function of time for different constant temperatures. For approximate estimation of the influence of temperature on the aging process, we can consider that increase of the temperature by 10° C doubles the aging rate and, accordingly, halves the seal service life [34]. The aging coefficient k_{ag}, characterizing the aging rate change with temperature increase by 10° C, varies from 1.3 to 2.4 for different rubbers and different temperatures. The exact values of k_{ag} can be found from handbooks on technical rubber products.

Aging of the other hydraulic drive structural components usually has negligibly small influence on hydraulic drive serviceability and characteristics.

For the servocontrols during the process of accumulating operating time, there is aging of the magnetic and electrical insulation materials; however, their service life, as a rule, exceeds by an order of magnitude the service life of the rubber seals (in terms of aging).

A third destructive factor is fatigue damage accumulation in hydraulic drive power components under the action of variable loads. The fatigue damage accumulation process in hydraulic drive components was examined in Section 6.1. We note that during hydraulic drive service life tests, fatigue damage accumulates primarily in certain components, while during vibration tests the damage accumulates in other components. While during vibration tests the highest

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loads are experienced by the structural components with low resonant frequencies and also the hydraulic drive mounts, during service life tests the highly-loaded output components (piston, piston rod, link), actuator housing, distribution valve housing, and hydraulic drive mounts experience the highest loads (although the nature of the mount loading is different in these cases).

In many cases, the structural components are subjected during service life tests to the simultaneous action of varying and constant loads. For example, all the components for which fatigue damage accumulation is due to pressure changes (actuator housing, distribution valve housing, and so on) are subjected to such loading. In this case, the variable stress with amplitude σ and the constant stress σ_0 are reduced to an equivalent symmetric variable stress $\sigma_e = \sigma + \alpha_0 \sigma_0$, where the coefficient of reduction of the asymmetric loading cycle to the symmetric cycle ($\alpha_0 = 1 - \sigma_T/\sigma_u$) depends on the material and is, on the average, equal to about 0.2. Although aircraft hydraulic drives are also subject during operation to cyclic temperature variation, the temperature change gradients are relatively small and usually do not lead to significant thermal stresses. Moreover, for properly designed hydraulic drives with mating details made from materials with different thermal expansion coefficients, significant thermal stresses due to different deformation of the mating details do not arise with change of the temperature regime. Therefore, the questions of thermal strength do not influence hydraulic drive service life.

Characteristic for certain aircraft hydraulic drives is the "parking" regime with constant load on the stationary output member. In this case, the hydraulic drive structural components are subject to the long-term action of constant stresses at the operating temperature. The influence of such loads on structural strength is examined in stress-rupture strength theory. The resistance to stress-rupture fracture is characterized by the stress-rupture strength curve, yielding the relationship between the stress magnitude σ_{sr} and time T_{sr} required for fracture at this stress (Figure 8.3). /181

For constant temperature this relation, described using a logarithmic time scale by a straight line, may be expressed in the form:

$$T_{sr}=A_{sr}\sigma_{sr}^{-m_{sr}}, \qquad (8.10)$$

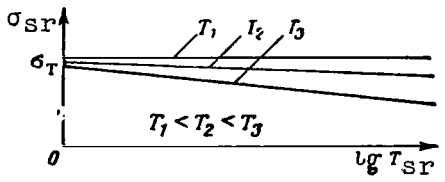

Figure 8.3. Typical stress-rupture strength curves

where A_{sr} and m_{sr} are positive quantities which are constant for given material and given temperature (m_{sr} = 10 - ∞).

The stress-rupture strength of a material is characterized by the function $T_{sr} = t\ (\sigma_{sr}, t°)$, or by a family of stress-rupture strength curves. The stress-rupture strength decreases with temperature increase, and the curves shown in Figure 8.3 shift downward relative to the σ_{sr} axis with simultaneous increase of their slope, which corresponds to decrease of the exponent m_{sr} in (8.10) with temperature increase.

For the materials usually used in aircraft hydraulic drives in the operational temperature range, stress-rupture strength has practically no influence. Therefore, in most cases, we can ignore the hydraulic drive "parking" regimes with constant load on the output member when formulating the program for hydraulic drive service life bench testing.

8.2. Hydraulic Drive Service Life Test Technique

Reliable hydraulic drive functioning in service is usually guaranteed by the results of bench service life tests. The service life test program is specified on the basis of analysis of the hydraulic drive operating regime in service. For stabilization system hydraulic drives, the service life test duration usually coincides with the service life established by these tests, since these hydraulic drives are in motion practically continuously during flight. Depending on the flight phase and flight conditions, the control /182

signals to the hydraulic drives, mechanical loads, and working fluid and ambient temperatures vary during each flight. However, in formulating the service life test program, the actual hydraulic drive operating regime cyclograms are averaged, and several long-duration regimes with constant values of the control signals, loads, and temperatures are specified. In each regime, the output member motion is sinusoidal, and the hydraulic drive output member amplitude and frequency are constant, and correspond to the operating regime. In certain cases, the operating time in the thermal regime is carried out in cycles with temperature variation from standard to maximal and back at a specified rate; in this case, each thermal cycle corresponds to a single flight.

For hydraulic drives which operate only in certain flight stages, service life test duration is less than the service life guaranteed by these tests, and the number of test cycles is equal to the number of operating cycles during the service life. Holding regimes with the output member stationary at elevated temperature, either with or without hydraulic pressure applied, may also be included in the hydraulic drive service life test program.

Service life tests are conducted on conventional test stands or thermal stands. The controlling signal for servocontrols is applied from a low-frequency periodic signal generator, and for boosters — from a sine generator. A cycle-completed counter must be provided on the stand used for service life testing. If it is necessary to carry out simultaneously acceptance-delivery and inspection tests of several hydraulic drives, it is advisable to have a separate simplified test stand for the long-duration service life tests. It is recommended that a variable-output pump be installed in the hydraulic supply system of the stand intended for service life tests. /183

Normally, the hydraulic drive goes through a whole series of required checks under normal conditions, and also with climatic and mechanical inputs prior to beginning the service life tests. After carrying out the service life tests, a recheck should be made of the hydraulic drive parameters which are affected by the accumulation of

irreversible changes which take place during service life testing. After completing the life service tests, inspection checks should be made under normal conditions, and also tests for heat stability and cold stability. The scope of the inspection tests following the service life tests can be reduced, since checks of several functional and structural parameters need not be made (for example, after service life tests, we do not need to check the weight, output member travel, dynamic characteristics, and so on).

As a rule, in experimental production and in the beginning of series production, service life tests are run on the basis of limited initial service life, less than the required guaranteed service life. Further service life increase is carried out either by stage-by-stage increase of the life on lead-airplanes, or by long-term test-stand normal or accelerated service life tests.

In accordance with the method of stage-by-stage increase of the initial service life using lead-airplanes, 10 - 20 hydraulic units are installed aboard a group of airplanes specially assigned for this purpose. After the units have operated in flight for the initial service life, they are taken from the airplane and some (2 - 4) of them are sent to the manufacturer. Half of the units received at the manufacturer's plant are torn down, and the other half are subjected to additional test-stand service life tests, after which the units which have gone through this testing are also torn down. If the results of these tests are positive, and defect analysis of the parts of the units which are torn down is positive, an increased service life for the units is established, equal to the sum of the initial and additional service lives. After this, operation of the units on the lead-airplanes is continued until the increased service life is run out. Further increase of the service life proceeds similarly. /184

This technique for increasing service life provides good validity of the results obtained, since the influence of the difference between the test stand service life test regimes and the operational regimes is reduced in comparison with test stand service life tests

for the complete service life. However, this technique for establishing service life has drawbacks, which involve the necessity for a long calendar time interval for increasing the service life to the required value and, consequently, the absence in the development and initial series production stage of information on serviceability of the unit for the complete specified service life. Moreover, it is not always possible to use lead-airplanes. Therefore, the question of developing accelerated service life testing methods becomes very important.

8.3. Techniques for Accelerating Hydraulic Drive Service Life Testing

In connection with the rapid development of aviation and the considerable increase of the required service lives, the test stand service life tests have become an obstacle in programs to develop long-life hydraulic drives. This is due to the fact that the time from the moment of initial hydraulic drive development to their entry into series production has shortened markedly, and also to the long duration of the stand tests. A way out of this situation is the conduct of accelerated service life tests. In so doing, in order to obtain reliable data on hydraulic drive operating life, it is necessary to establish analytic relationships between the hydraulic drive regimes and parameters during normal and accelerated tests. The importance of the accelerated service life test methods is very great, since they are the key to obtaining timely and reliable data on hydraulic drive operating life.

Accelerated service life tests are considered to be any tests which permit obtaining information on the operating life of products in less time than during tests in the operational regimes. The effectiveness of accelerated service life tests can be characterized by the acceleration coefficient (k_a), equal to the ratio of the running time under normal operating conditions (T_n) to the running time

in the accelerated tests (T_a), which validate the same service life: /185

$$k_a = \frac{T_n}{T_o} .$$

The Soviet and foreign literature contains general information on the possibility of conducting accelerated service life tests of units in the forced regime and accelerated service life tests in the normal (operating) regime; however, no concrete data are available on the techniques for accelerated tests.

Let us examine the basic techniques for accelerating service life tests. The simplest way to accelerate system service life tests is to conduct tests under conditions in which the system operates more intensely in time (without interruptions and no-load regimes) than during tests in the normal regime, which shortens the test time correspondingly. For example, we can exclude regimes in which the hydraulic drives are under pressure in the absence of control signals and with normal working fluid and ambient temperatures. This technique can be used to accelerate the tests of hydraulic drives which operate intermittently during flight. However, this very simple acceleration technique can be used only after analysis of the influence of interruptions in system operation on the processes of irreversible change accumulation during the service life. If all the no-load regimes (T_{nl}) are excluded and tests are conducted only in the operating regimes (T_o), then $k_a = 1 + T_{nl}/T_o$.

Sometimes, accelerated tests based on the prediction method are used. In this case, the hydraulic drive is tested in the operating regime for a limited time (less than the service life validated by these tests), and on the basis of system condition after the abbreviated tests, a conclusion is drawn on system serviceability for the complete required service life. In this case, we assume that the rate of the irreversible change accumulation process is constant, /186 and the value of this rate, determined during the abbreviated tests,

is used to estimate the system condition after the service life. However, the possibilities of this accelerated test technique are very limited for hydraulic drives, since it is practically impossible to make an adequately accurate quantitative estimate after abbreviated tests of the degree of wear, aging, and fatigue damage accumulations. Moreover, establishing the limits of the final service life of the parts and components presents considerable difficulty, including the necessity for repeated teardown of the test hydraulic drives in order to monitor the irreversible change accumulation process, which then requires additional running. The prediction technique is completely inapplicable in experimental production, since it requires first obtaining a large amount of statistical data on irreversible change accumulation behavior in the process of building up the operating time.

It appears that the most promising technique for hydraulic drives is to conduct accelerated service life tests in the forced regimes. The forced accelerated service life test regimes must be equivalent to the normal service life tests (i.e., tests in the operational regimes).

The service life test regimes can be considered equivalent if the condition of the hydraulic drive (and also its parts and components) and the change of the average magnitudes of the hydraulic drive parameters are the same after the normal and the accelerated tests. It is obviously impossible to achieve complete equivalence of the normal and accelerated test regimes in regard to all the components and parameters.

As was shown in Section 8.1, the basic destructive factors for hydraulic drives are wear, aging, and fatigue damage accumulation. In developing accelerated service life test techniques, it is necessary to ensure equivalence of the normal and accelerated tests with regard to these destructive factors.

However, hydraulic drives are very complex systems with a multitude of structural components of various types. Since the laws

governing the change of the rate of wear or aging during accelerated tests for the different structural components may differ significantly, it is necessary to select for each of the destructive factors one or more hydraulic drive components which are critical (weak) in regard to the given factor (see Figure 8.1). /187

On the basis of analysis of hydraulic drive operation and test results, and also studies of the properties of the structural components of hydraulic drives with translating output member, when operating for the guaranteed service life, the critical components with respect to wear are the piston rod seals; with respect to aging — the rubber seals; and with respect to fatigue damage accumulation — the structural components of the piston group and the cylinder housing.

The basic operating regime forcing techniques are increase of output member velocity and increase of working fluid temperature. Increase of the output member velocity with sinusoidal motion law can be accomplished by increasing the frequency with constant output member displacement amplitude, or simultaneous increase of the frequency and amplitude. Usually, the output member velocity in the operating regimes is less than the maximal available hydraulic drive velocity, which makes it possible to increase the output member motion frequency (or amplitude) by several fold. In the accelerated tests, it is necessary to force the regimes so as to ensure equivalence in regard to the destructive factors for the critical components.

In selecting the forced regimes, we must consider the limitations on maximal output member velocity, maximal permissible working fluid and seal temperature, maximal load on the output member. It is necessary that there be no qualitative changes of the irreversible change accumulation processes when forcing the regimes. Usually, no qualitative changes of the wear, aging, and fatigue damage accumulation processes are observed in the operating range of hydraulic drive parameter variation.

In analyzing the accelerated service life test regimes, we must be certain that after accelerated tests in the forced regimes for the time T_{ga}, the average values of the wear, aging, and fatigue damage accumulation of the critical components are the same as in normal tests after the time T_g. The variations of the irreversible change accumulation rate (L_{1a}) and the failure distrubution density $f_{1a}(t)$ for the weakest component in the accelerated tests are shown in Figure 8.1. For this test acceleration technique, we can consider /188 that the reduction of guaranteed service life test duration is proportional to the reduction of the mean time between failures T_{0a}:

$$T_{ga} = T_g \frac{T_{0a}}{T_0}$$

Using the condition of constancy of the amount of eroded material in the normal and accelerated tests ($\Delta V_n = \Delta V_a$), and considering that the actual contact areas are equal ($S_{an} = S_{aa}$), we can obtain with the aid of (8.3) the following equivalence condition, based on wear:

$$i_n L_n = i_a y_a,$$

where L_n and L_a are the friction paths in the normal and accelerated tests.

For the case of acceleration with respect to frequency with sinusoidal output member motion in regimes which are equivalent in regard to wear, the connection between the number of cycles (n_n and n_a) and test time in the normal and accelerated tests (τ_n and τ_a) has the form:

$$n_a = n_n \frac{i_n}{i_a}, \qquad (8.12)$$

$$\tau_a = \tau_n \frac{i_n}{i_a} \frac{v_n}{v_a}, \qquad (8.13)$$

where v_n and v_a are the maximal output member velocities during the cycle.

Considering the expressions (8.5) - (8.8), we can obtain:

$$i_a = i_n \left(\frac{v_a}{v_n}\right)^{-ab} \left(\frac{t_n = \Delta t_k}{t_n}\right)^{a_1} \text{ for } t_n - t_a, \qquad (8.14)$$

$$i_a = i_n \left(\frac{t_a}{t_n}\right)^{a_1} \text{ for } v_n = v_a, \qquad (8.15)$$

where $\Delta t_k = k_t (v_a - v_n)$.

In the case of simultaneous forcing of the tests with respect to output member velocity and working fluid temperature, the dependence of the number of cycles and test time for the normal and accelerated tests can be represented in the following form, considering (8.12) — (8.15): /189

$$\tau_a = \tau_n \left(\frac{v_n}{v_a}\right)^{1-ab} \left(\frac{t_n}{t_a + \Delta t_k}\right)^{a_1}, \qquad (8.16)$$

$$n_a = n_n \left(\frac{v_n}{v_a}\right)^{-ab} \left(\frac{t_n}{t_a + \Delta t_k}\right)^{a_1}. \qquad (8.17)$$

Figures 8.4 and 8.5 show in dimensionless form the curves of test time and number of cycles as functions of the degree of forcing with respect to velocity and temperature on the basis of wear equivalence.

When evaluating the equivalence of the normal and accelerated tests with respect to thermal aging, we took as the basis the same

variation of the seal material physical and chemical properties during the thermal aging process:

$$u_n \tau_n^{ag} = u_a \tau_a^{ag}, \qquad (8.18)$$

where τ_n^{ag} and τ_a^{ag} are the rubber seal aging times for the normal and accelerated tests; u_n and u_a are the seal material oxidation process rates for the normal and accelerated tests.

After substituting (8.9) into (8.18), we obtain the following connection between the seal material aging times τ_n^{ag} and τ_a^{ag} and the parameters of the material property physical and chemical change processes:

$$\tau_a^{ag} = \tau_n^{ag} e^{\frac{1}{R}\left(\frac{E_a}{T_a} - \frac{E_n}{T_n}\right)}. \qquad (8.19)$$

$k_a = \tau_n/\tau_a$

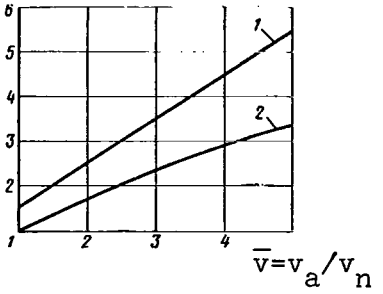

Figure 8.4. Wear acceleration versus output member velocity increase (for constant amplitude):
1 — $t_n = t_a$; 2 — t_n = 100° C; t_a = 120° C

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$n = n_a/n_n$

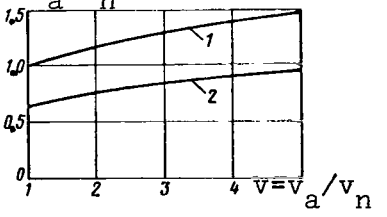

Figure 8.5. Number of cycles versus output member velocity (for wear equivalence):
1 — $t_n = t_a$; 2 — t_n = 100° C; t_a = 120° C

However, since a large amount of experimental data has now been accumulated on seal material service life variation as a function of temperature, it is convenient to represent (8.19) in the form:

$$\tau_a^{ag} = \tau_n^{ag} k_{ag}^{-0.1 (t_a - t_n + \Delta t_k)}. \qquad (8.20)$$

The dependence of the aging acceleration factor ($k_a^{ag} = \tau_n^{ag}/\tau_a^{ag}$) on the degree of temperature increase for different aging coefficients is shown in Figure 8.6.

When the number of cycles in the accelerated regime is not equal to the number of normal regime cycles, in order to maintain equivalence with respect to fatigue damage accumulation, we must alter the stresses in the structural components (or output member load amplitude) in accordance with the following expressions: /191

$$\sigma_a = \sigma_n \left(\frac{n_n}{n_a}\right)^{1/m},$$

$$R_a = R_n \left(\frac{n_n}{n_a}\right)^{1/m}, \qquad (8.21)$$

where the quantity m may be taken equal to 5.

Figure 8.7 shows the amplitude of the cyclically varying load on the output member as a function of the ratio of the number of cycles with satisfaction of the equivalence conditions with respect to fatigue damage accumulation.

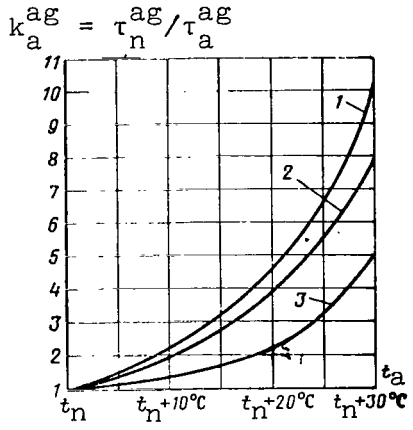

Figure 8.6. Aging acceleration factor versus temperature increase:
1 — k_{ag} = 2.2; 2 — k_{ag} = 2; 3 — k_{ag} = 1.5

In the case when the tests are forced with respect to hydraulic drive output member velocity, we observe increase of the working fluid leakage through the moving joint seals, which agrees well with the theoretical data of [34]. Therefore, increased working fluid leakage through the moving joint seals is permitted in the accelerated regimes. If, in the accelerated regimes, leaks are found which exceed the allowable leakage for the normal regime, a check of the leakage in the normal regime must be made after completing the service life run in the given accelerated regime.

The relations presented in Chapter 8 make it possible to calculate the accelerated test regime for which equivalence with respect to wear, aging, and weak component fatigue damage accumulation is achieved simultaneously. To this end, as a result of analysis of the normal service life test regime parameters and also the maximal possible output member velocities and working fluid temperatures for test hydraulic drive, we select the parameters of the regime which is forced with respect to velocity and temperature and, on the basis of (8.16) and (8.17), we determine the number n_a of cycles required for satisfaction of equivalence with respect to wear (for constant output member velocity) and the test time τ_a. Then we use (8.20) to find τ_a^{ag}. If $\tau_a \neq \tau_a^{ag}$, by varying the degree of forcing with respect to velocity and temperature, we obtain by successive approximations the accelerated test regime parameters for which equivalence with respect to wear and aging is ensured during the test time $\tau_a = \tau_a^{ag}$. When forcing several specified regimes, it is sometimes more convenient to evaluate only the overall equivalence with regard to aging or wear for all the regimes. In the calculation, $\pm$ 5% deviation from the equivalence condition is permitted. If $n_a \neq n_n$, then in order to maintain equivalence with respect to fatigue damage accumulation, we must alter the amplitude of the alternating load on the output member in accordance with (8.21). Usually the required load change is slight, and its influence on wear can be neglected.

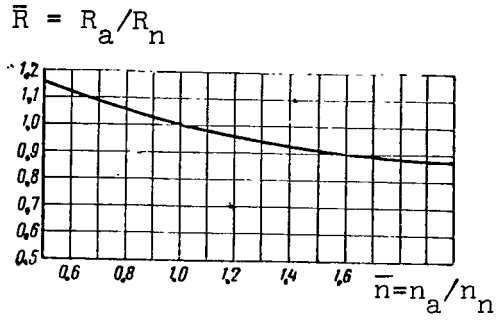

Figure 8.7. Output member load amplitude versus ratio of number of cycles (for equivalence with respect to fatigue damage accumulation)

In these computational relations, we assume that the output member sinusoidal motion amplitude is the same in the normal and accelerated tests. In this approach, the nature of the output member moving seal wear is as similar as possible in the normal and

accelerated tests. However, in those cases when the number of cycles n_a differs markedly from n_n, and it is necessary to alter the load amplitude by more than 10%, it is advisable to change the output member motion amplitude in the forced regime so that $n_a \approx n_n$. In those cases when questions of fatigue strength in the service life tests are most critical, we should ensure equality of the number of cycles and output member load amplitude in the accelerated and normal regimes, and the equivalence with respect to wear and aging should be /193 provided by proper selection of the output member frequency (f_a) and displacement amplitude (A_a).

Equating the accelerated test times $\tau_a = \tau_a^{ag}$, we can obtain the following computational relations for the subject acceleration technique (i.e., for $n_n = n_a$):

$$f_a = f_n k_{ag}^{0.1 (t_a - t_n + \Delta t_k)},$$

$$\left(\frac{A_a}{A_n}\right)^{1+ab} = \left(\frac{f_n}{f_a}\right)^{ab} \left(\frac{t_n}{t_a + \Delta t_k}\right)^{a_1}.$$

The proposed technique for accelerating service life testing makes it possible to obtain an acceleration factor $k_a = 3 - 5$, and, in certain cases, k_a may reach 10 - 20.

Accelerated service life tests can be used extensively in comparative tests of several design versions, and for determining the weak hydraulic drive structural members with respect to fatigue strength, wear resistance, corrosion resistance, and so on. In this case, the task of creating an accelerated test technique simplifies considerably, since complete equivalence of the normal and accelerated tests is not required. In the case of tests for wear resistance of moving joint seals, the basic test forcing techniques involve increasing the frequency and amplitude of the output member motions and increasing the working fluid temperature. Increase of the ambient air temperature has practically no effect on the seal operating

conditions. Acceleration of tests for wear resistance or fatigue strength can also be accomplished by increasing the load on the hydraulic drive output member and simultaneous increase of the supply pressure in the hydraulic supply system. For an accelerated estimate of the wear resistance of the hydraulic drive internal components and its distribution valve, a working fluid with high degree of contamination can be used. To this same end, tests can be conducted using a working fluid with poor wear resistance properties — kerosene, for example. For accelerated evaluation of hydraulic drive internal chamber corrosion resistance, it is advisable to use a working fluid with increase of the moisture content to 2 - 5%. /194
Sometimes a combination of several of the aforementioned accelerating techniques is used.

8.4. Determination of Reliability Indices From Test Results

The conduct of special tests to determine quantitative reliability indices for hydraulic drives is impossible in practice. This is due to the high requirements on hydraulic drive reliability (for example, P = 0.999 or more for four hours of operation), which requires a very large volume of tests, and also because of the long duration and cost of hydraulic drive testing and the small-scale nature of their production. The conduct of hydraulic drive reliability testing in the development and experimental production stage is even less practical.

If necessary, the results of bench inspection tests and flight operation data can be used for approximate determination of hydraulic drive reliability in the experimental and series production stages. The confidence interval method is most convenient for evaluating hydraulic drive reliability. Specifying the confidence level P*, we can at any stage find, from the available data on total hydraulic drive operating time T_Σ and number of failures m_Σ during this operating time, the mean operating time to failure $T_0 = T_\Sigma/m_\Sigma$ and also the upper and lower confidence limits T_{0u} and T_{0l} for the mean

operating time and the corresponding failure rates λ, λ_u, and λ_l [44]. Then we can find from the λ values, the magnitudes of the confidence limits of the probability of reliable operation for the specified time (usually for the time of flight). If there were no failures in the initial operating time data ($m_\Sigma = 0$), the upper confidence limit for the mean operating time is equal to infinity, and only the lower confidence limit is evaluated. The objective of the hydraulic drive reliability estimates made in this fashion is to make certain that the lower confidence limit $P_l(t_0)$ exceeds the specified minimal value $P(t_0)$. When applying this method in the experimental production stage, we use the data on operating time during inspection /195
tests, and here failures whose cause is established and to eliminate which effective methods have been taken may be ignored.

Considering the difficulties in confirming the specified hydraulic drive reliability indices and the increase of their reliability in the course of development and production, it is advisable to establish different reliability requirements for the development and initial series production stage and for the established series production stage (3 - 5 years after initiation of operation). It is advisable to establish the required reliable operation probability level for the first stage approximately an order of magnitude lower than for the second stage. In the first stage, the confidence level P* should be taken equal to 0.6, and in the second stage — 0.9.

In estimating the probability of reliable hydraulic drive operation in flight, we should consider only those failures which may lead to failure of the airplane to perform its flight mission. All forms of failures which lead to premature hydraulic drive removal are taken into account in evaluating the reliability and maintainability indices. The identification among all the basic possible hydraulic drive failure modes of those failure modes which can lead to failure to perform the flight mission is made by careful analysis of the influence of hydraulic drive characteristic changes on airplane flight control system operations.

Thus, the primary sources of information for evaluating aircraft hydraulic drive reliability are the results of bench tests (primarily service life tests) and operation. For comparatively simple hydraulic units and, particularly, for standardized components, the possibility of conducting special tests to determine the reliability indices becomes realistic as a result of the development and introduciton of effective accelerated testing methods.

References

1. Akhmatov, A. S. Molekulyarnaya fizika granichnogo treniya (Molecular Physics of Boundary Friction). Moscow, Fizmatgiz Press, 1963. /196

2. Bashta, T. M. Gidravlicheskiye sledyashchiye privody (Hydraulic Servo Drives). Moscow, Mashgiz Press, 1961.

3. Bashta, T. M. Gidravlicheskiye privody letatel'nykh apparatov (Aircraft Hydraulic Drives). Moscow, Mashinostroyeniye Press, 1967.

4. Belevitin, B. V., I. M. Krassov and M. P. Selivanov. Control System Electrohydraulic Servomechanism Redundancy. Pribory i systemy upravleniya, No. 2, 1970.

5. Belyanin, P. N. and Zh. S. Chernenko. Aviatsionnye fil'try i ochistiteli gidravlicheskikh sistem (Aircraft Filters and Hydraulic System Purifiers). Moscow, Mashinostroyeniye Press, 1964.

6. Berdichevskiy, B. Ye. Otsenka nadezhnosti Apparatury Avtomatiki (Evaluating Automatic Control Equipment Reliability). Moscow, Mashinostroyeniye Press, 1966.

7. Borisov, V. N. On the Mechanism of Seal Wear. Izvestiya Vuzov SSSR, Mashinostroyeniye, No. 2, 1969.

8. Bulovskiy, P. I. and E. M. Idel'son. Ispytaniya aviatsionnykh priborov (Aircraft Instrument Testing). Moscow, Mashinostroyeniye Press, 1966.

9. Vlasov-Vlasyuk, O. B. Eksperimental'nye metody v avtomatike (Experimental Methods in Automatic Control). Moscow, Mashinostroyeniye Press, 1969.

10. Gamynin, N. S. Osnovy sledyashchego gidravlicheskogo privoda (Hydraulic Servo Drive Fundamentals). Moscow, Oborongiz Press, 1962.

11. Gidravlicheskiy sledyashchiy privod (Hydraulic Servo Drive). Ed. V. A. Leshchenko. Moscow, Mashinostroyeniye Press, 1968.

12. Gidroagregaty dlya promyshlennogo primeneniya (Hydraulic Units for Industrial Use), Catalog. Moscow, Mashinostroyeniye Press, 1966.

13. Guillon, M. Issledovaniye i raschet gidravlicheskikh sistem (Hydraulic System Investigation and Calculation). Moscow, Mashinostroyeniye Press, 1964.

14. Gudkov, A. I. and P. S. Leshakov. Vneshniye nagruzki i prochnost' letatel'nykh apparatov (Flight Vehicle External Loads and Strength). Moscow, Mashinostroyeniye Press, 1968.

15. Dammer, A. and B. Griffin. Ispytaniya radioelektronnoy apparatury i materialov na vozdeystviye klimaticheskikh i mekhanicheskikh uslovii (Testing of Electronic Equipment and Materials Under Climatic and Mechanical Conditions). Moscow-Leningrad, Energiya Press, 1965.

16. Ivanova, V. S. Ustalostnoye razrusheniye materialov (Fatigue Failure of Materials). Moscow, Mashgiz Press, 1963.

17. Iorish, Yu. I. Vibrometriya (Vibration Measurement). Moscow, Mashinostroyeniye Press, 1963.

18. Karpushin, V. B. Vibratsiya i udary v radioapparature (Vibration and Shock in Electronic Equipment). Moscow, Soviet Radio Press, 1971.

19. Komarov, A. A. Nadezhnost gidravlicheskikh sistem (Hydraulic System Reliability). Moscow, Mashinostroyeniye Press, 1969.

20. Kragel'skiy, I. V. Treniye i iznos (Friction and Wear). Moscow, Mashinostroyeniye Press, 1968. /197

21. Kragel'skiy, I. V. and Ye. F. Nepomnyashchiy. On the Fatigue Wear Mechanism with Elastic Contact. Press of the Academy of Sciences USSR, Mekhanika i mashinostroyeniye, No. 5, 1963.

22. Lozovskiy, V. N. Nadezhnost i dolgovechnost zolotnikovykh i plunzhernykh par (Reliability and Life of Valve and Plunger Pairs). Moscow, Mashinostroyeniye Press, 1971.

23. Luzanova, I. A., V. G. Neyman and V. N. Prokof'yev. Influence of Hydraulic Drive Elastic Property Variability on Hydraulic Motor Dynamics. Avtomatika i telemekhanika, No. 3, 1969.

24. Lewis, E. E. and H. Stern. Gidravlicheskiye sistemi upravleniya (Design of Hydraulic Control Systems),Moscow, Mir Press, 1966.

25. Malinskiy, V. D. Kontrol' i ispytaniya radioapparatury (Electronic Equipment Inspection and Test). Moscow, Energiya Press, 1970.

26. Mikheyev, M. A. Osnovy teploperedachi (Fundamentals of Heat Transfer). Moscow, Gosenergoizdat Press, 1956.

27. Neyman, V. G. Static and Dynamic Characteristics of Hydraulic Drive with Variable Output Pump. Izvestiya Vuzov SSSR, Moscow, Mashinostroyeniye Press, No. 7, 1966.

28. Nikitin, G. A. and A. A. Komarov. Raspredelitel'nye i reguliruyushchiye ustroystva gidrosistem (Hydraulic System Distributing and Regulating Devices). Moscow, Mashinostroyeniye Press, 1965.

29. Nikitin, G. A. and S. V. Chirkov. Vliyaniye zagryaznennosti zhidkosti na nadezhnost' raboty gidrosistemy letatel'nykh apparatov (Influence of Fluid Contamination on Reliability of Aircraft Hydraulic System Operation). Moscow, Transport Press, 1969.

30. Nosov, Yu. A. Influence of Low Temperatures on Seal Efficiency. Trudy Moscow Aviation Institute, Oborongiz Press, No. 117, 1959.

31. Ponomarenko, Yu. F. Ispytaniye gidroperedach (Hydraulic Transmission Testing). Moscow, Mashinostroyeniye Press, 1969.

32. Popov, E. P. and I. L. Pal'tov. Priblizhennye metody issledovaniya nelineynykh avtomaticheskikh sistem (Approximate Methods of Nonlinear Automatic System Analysis). Moscow, Fizmatgiz Press, 1960.

33. Potapov, A. M. Nastroyka i ispytaniya sledyashchikh privodov (Servo Drive Adjustment and Testing). Leningrad, Energiya Press, 1970.

34. Problems of Modern Sealing Technology. "Collection of Papers Presented at International Conference at Cranfield, Edited by V. N. Prokof'yev and L. N. Kondakov," Moscow, Mir Press, 1967.

35. Prokof'yev, V. N. Small Auto-Oscillations when Operating at Creeping Velocities. Avtomatika i telemekhanika, no. 5., 1968.

36. Razdolin, M. V. Uplotneniya aviatsionnykh gidravlisheskikh agregatov (Sealing of Aircraft Hydraulic Systems). Moscow, Mashinostroyeniye Press, 1965.

37. Raschety na prochnost' v mashinostroyenii (Strength Calculations in Machine Design) edited by S. D. Ponomarev. Vol. 3. Moscow, Mashgiz Press, 1959.

38. Rumshinskiy, L. Z. Matematicheskaya obrabotka rezul'tatov eksperimenta (Mathematical Analysis of Experimental Results). Moscow, Nauka Press, 1971.

39. Rybalov, S. L. On the Nature of Rubber Wear and Failure During Friction on a Metallic Surface at High Temperatures. In the book: Voprosy Treniya i Problemy Smazki (Friction and Lubrication). Press of the Academy of Sciences USSR, 1968.

40. Sluchaynye kolebaniya (Random Vibrations), edited by S. H. Crandall. Moscow, Mir Press, 1967.

41. Sotskov, B. S. Osnovy teorii i raschety nadezhnosti elementov i ustroystv avtomatiki i vychislitel'noy tekhniki (Fundamentals of Theory and Design of Automatic Control and Computer Components and Systems). Moscow, Vysshaya Shkola Press, 1970.

42. Elektrogidravlicheskiye sledyashchiye sistemy (Electrohydraulic Servo Systems), edited by V. A. Khokhlov. Moscow, Mashinostroyeniye Press, 1971.

43. Krushchov, M. M. Classification of Machine Part Wear Conditions and Modes. In the book: Treniye i iznos v mashinakh (Friction and Wear in Machines). Press of the Academy of Sciences USSR, No. 8, 1953. /198

44. Shor, Ya. B. Staticheskiye metody analiza i kontrolya kachestva i nadezhnosti (Statistical Methods of Quality and Reliability Analysis and Control) Moscow, Soviet Radio Press, 1962.

45. Brook, R. H. W. and G. S. C. Parry. Cumulative Damage in Fatigue: A Step Towards its Understanding. Journal of Mechanical Engineering Science, Vol. 11, No. 3, 1969.

46. Cheng-Kuo, Weng. Transmission of Fluid Power by Pulsating-Flow Concept in Hydraulic Systems. Trans. of ASME, Ser. D., No. 2, 1966.

47. Lang, C. Elastic Seals — Pressing Behavior and Friction. Maschinenmarkt., No. 96, 1969, p. 75.

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